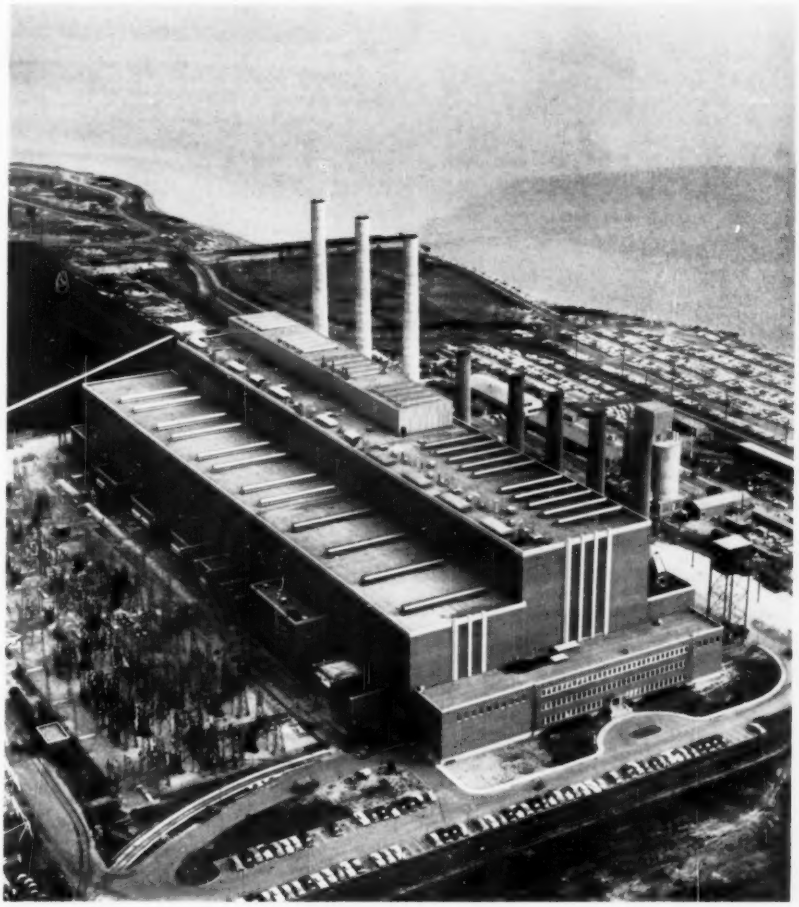
# Combustion

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION



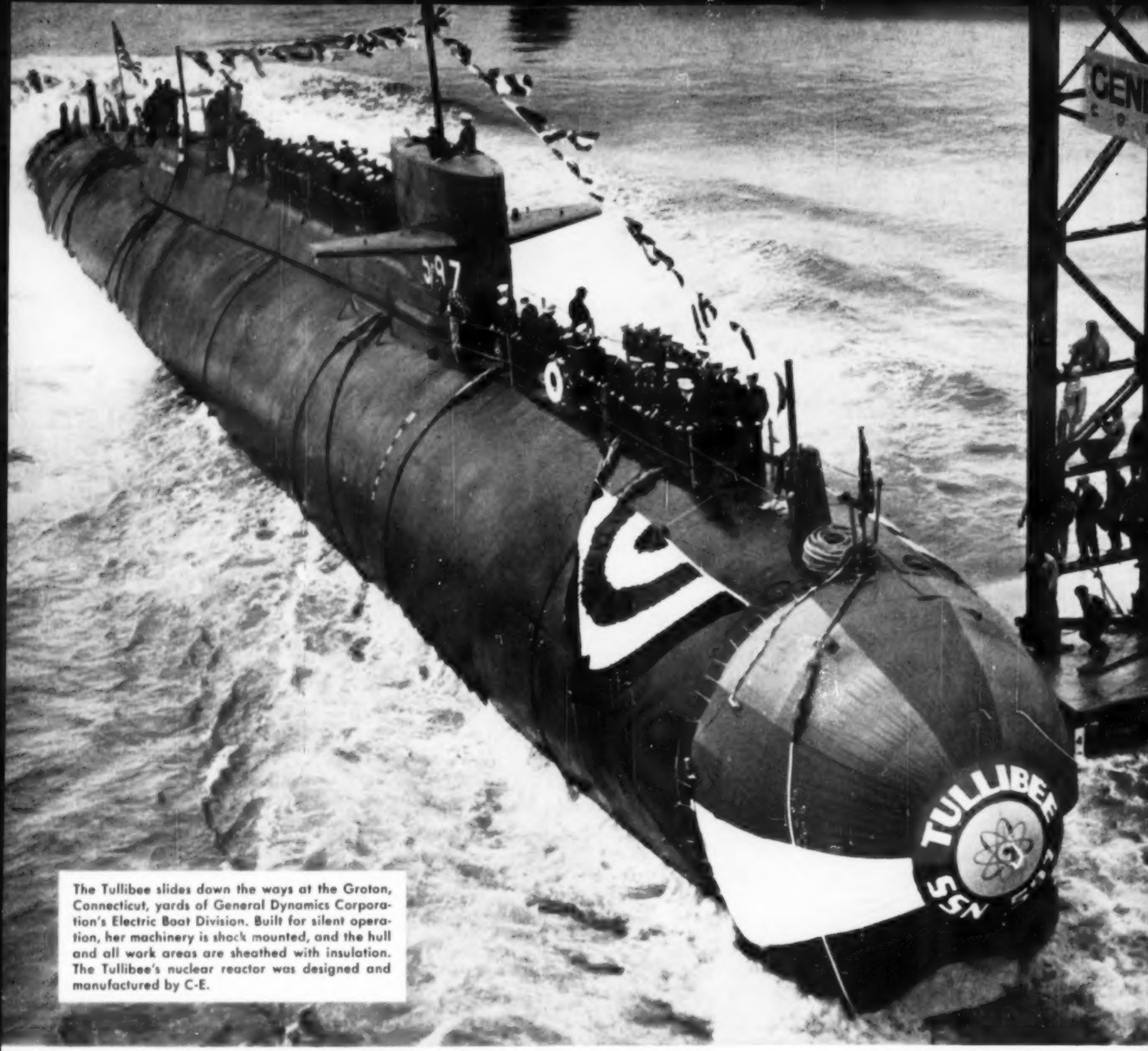
June 1960

*Economic Condenser Sizing*

*Steam Power Plant Clinic*

*American Power Conference*

*Prototype Gas Turbine Experiences*



The Tullibee slides down the ways at the Groton, Connecticut, yards of General Dynamics Corporation's Electric Boat Division. Built for silent operation, her machinery is shock mounted, and the hull and all work areas are sheathed with insulation. The Tullibee's nuclear reactor was designed and manufactured by C-E.

# THIS SUPER-SILENT HUNTER-KILLER HAS KEEN EARS

The Tullibee, the first nuclear powered "hunter-killer" submarine, launched on April 27th, is designed to find and destroy enemy submarines beneath the sea's surface. It is the first submarine in the nuclear Navy to have a turbo-electric drive. It utilizes the most advanced hull design, is packed with sonar equipment and represents a major advancement in the Navy's anti-submarine warfare development program.

Silence is vital to Tullibee's mission and, as a result, she's one of the quietest underseas craft ever built. She is also unique in that her torpedo tubes are located amidships rather than in the bow. This allows Tullibee's nose to be loaded with an unprecedented number of sonar tracking transducers and hydrophones.

Combustion Engineering designed the nuclear power plant of the Tullibee and manufactured most of its major components, including the core with its fuel element assemblies. A prototype installation, including a portion of a submarine hull, has been in full power operation for some time at the C-E Naval Reactor Division in Windsor, Connecticut. This prototype, also designed and built by C-E, is the only installation of its kind at a privately owned site. It is now used for test purposes and training Navy crews.

The Tullibee adds one more name to the long list of notable power installations designed and built by C-E . . . installations that are representative of the most advanced practice in steam generation on land or sea.

* * *

*C-E also designed and built the reactor vessels and steam generators for the USS Triton, world's largest submarine, which recently encircled the globe, submerged, in 83 days.*

**COMBUSTION ENGINEERING**

Combustion Engineering Building • 200 Madison Avenue, New York 16, N. Y.

C-279

ALL TYPES OF STEAM GENERATING, FUEL BURNING AND RELATED EQUIPMENT; NUCLEAR REACTORS; PAPER MILL EQUIPMENT; PULVERIZERS; FLASH DRYING SYSTEMS; PRESSURE VESSELS; SOIL PIPE

# Combustion

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

volume 31 number 12 **June 1960**

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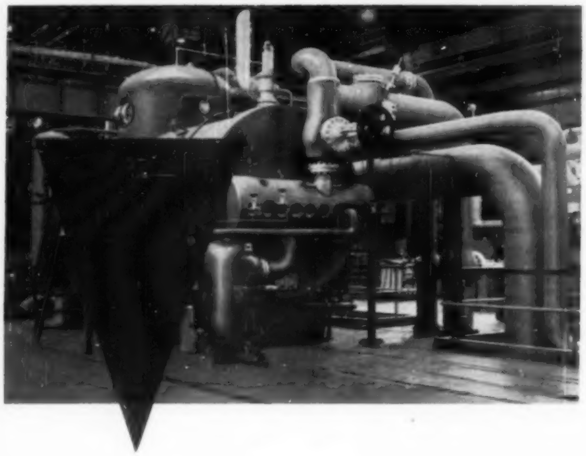
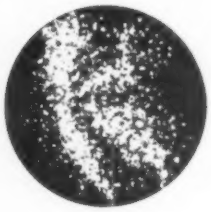
BPA

Printed in U. S. A.

*Three hundred foot high stacks mark the largest thermal-electric plant in Canada—Ontario Hydroelectric's Richard L. Hearn Station, Toronto*



# WHEN INDUSTRY NEEDS WATER



## PURER THAN THE DRIVEN SNOW

**Yuba Evaporators are achieving new record high purities**

*Other Yuba products for steam power plants include Condensers, Feedwater Heaters, Expansion Joints, Heaters, Tanks, Cranes, Structural Steel and scores of other items.*

In power, processing, marine or any industry that needs high purity water in volume, Yuba evaporators are recognized as the finest equipment available today. There are good reasons why. Recent tests by Consolidated Edison Company of New York show that purities better than 0.004 PPM total solids are achieved – purities "heretofore unknown." Heart of the Yuba evaporator is the patented improved bubble tray design. Yuba evaporators have reached their present peak because of continuous refinement of design, engineering and manufacturing. And Yuba's mechanical vapor purifier has also been found to be the most efficient of its type within the limits of mechanical purifier design.

Bubble tray or mechanical evaporators, Yuba designs are the most flexible in the industry. They can be installed vertically or horizontally—in a wide capacity range – depending on the industry need or basic plant design. Whatever your high purity water need, remember Yuba for the finest in evaporators. Write today for the complete story of tests on Yuba equipment – Bulletin YHT 101.

YUBA

*specialists in heat transfer equipment*

**YUBA HEAT TRANSFER DIVISION**

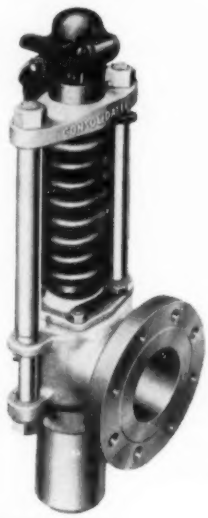
*4th and Main Streets, Honesdale, Pennsylvania*

**YUBA CONSOLIDATED INDUSTRIES, INC.**

Sales Offices in Atlanta • Buffalo • Chicago • Cleveland • Houston • Los Angeles • New York • Philadelphia • Pittsburgh • San Francisco • Seattle

# greater

**CONSOLIDATED "MAXIFLOW" SAFETY VALVES set a new standard of capacity and tightness in high pressure steam service**



*Consolidated "Maxiflow" Safety Valve, Type 1700 Series. Sizes: 1½" through 6". Pressure to 3000 psi. Temperatures to 1120° F. for superheater and reheater service.*

# capacity

Consolidated "Maxiflow" Safety Valves have greater discharge capacity, seat tightness, and shorter blowdown essential to optimum service on today's high-pressure, high-temperature steam generating equipment.

The highest degree of valve tightness is achieved with the Thermodisc Seat, an exclusive Consolidated feature. This unique seating design permits rapid equalization of temperature differentials set up when a high pressure valve reseats after blowing. The Thermodisc minimizes thermal stresses which cause seat distortion. Result: permanent tightness that saves on steam and valve maintenance.

The Consolidated "Maxiflow" is designed to function with a minimum blowdown. Blowdown adjustment can be made with the valve under full pressure.

Bulletin 707B contains complete details about Consolidated "Maxiflow" Safety Valves. Write for a copy.

MAXWELL MANNING M&M & MOORE INC. TRADE MARK

***CONSOLIDATED SAFETY VALVES***

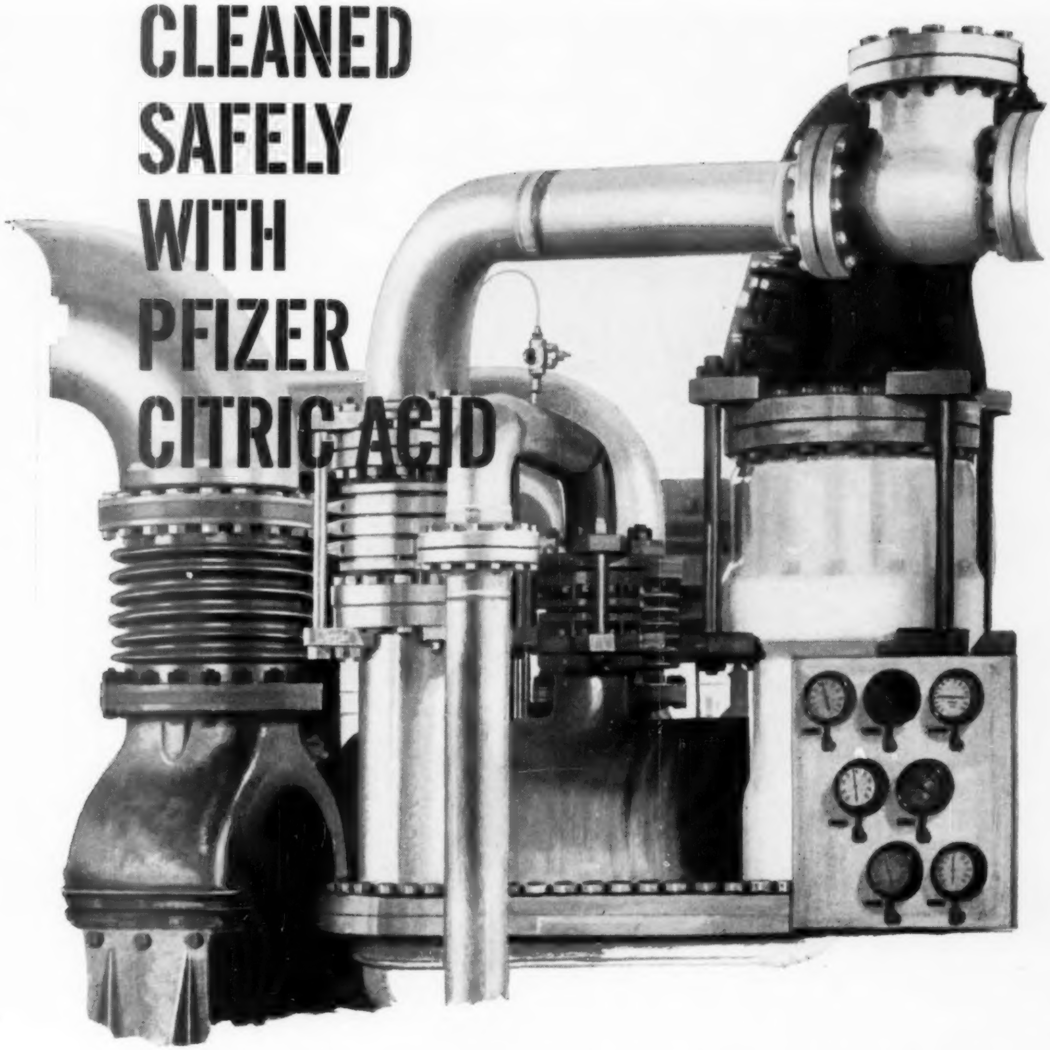
***A product of***

**MANNING, MAXWELL & MOORE, INC.**

*Valve Division • Stratford, Connecticut*

*In Canada: Manning, Maxwell & Moore of Canada, Ltd., Galt, Ontario*

# CLEANED SAFELY WITH PFIZER CITRIC ACID



STAINLESS STEEL BOILERS, HEAT EXCHANGERS, ATOMIC INSTALLATIONS, CHEMICAL PROCESSING EQUIPMENT CLEANED SAFELY, EFFICIENTLY WITH PFIZER CITRIC ACID

● Industry experience proves that citric acid eliminates chloride stress corrosion problems — provides effective descaling — permits easier, more efficient after-rinsing.

Discuss with your chemical cleaning service company these advantages of Pfizer Citric Acid in stainless steel cleaning solutions:

**1** Citric acid is highly efficient in removing imbedded metal and oxide films from stainless steel.

**2** Citric acid's excellent sequestering ability prevents reprecipitation of dissolved scale.

**3** Citric acid cleaning eliminates the problem of chloride stress corrosion.

**4** Citric acid can be effectively inhibited without losing its cleaning or sequestering ability.

**5** Citric acid is sold as a dry, 100% acid — meaning savings in storage and handling.

**6** Citric acid is water soluble, easy to handle, and non-toxic.

Let us send you further information, cost and obligation-free!

I want to learn more about the use of Pfizer Citric Acid for cleaning stainless steel equipment. Please send me Technical Bulletin 102.

Name____________________

Company____________________

Address____________________

City________ Zone____ State______

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Pfizer

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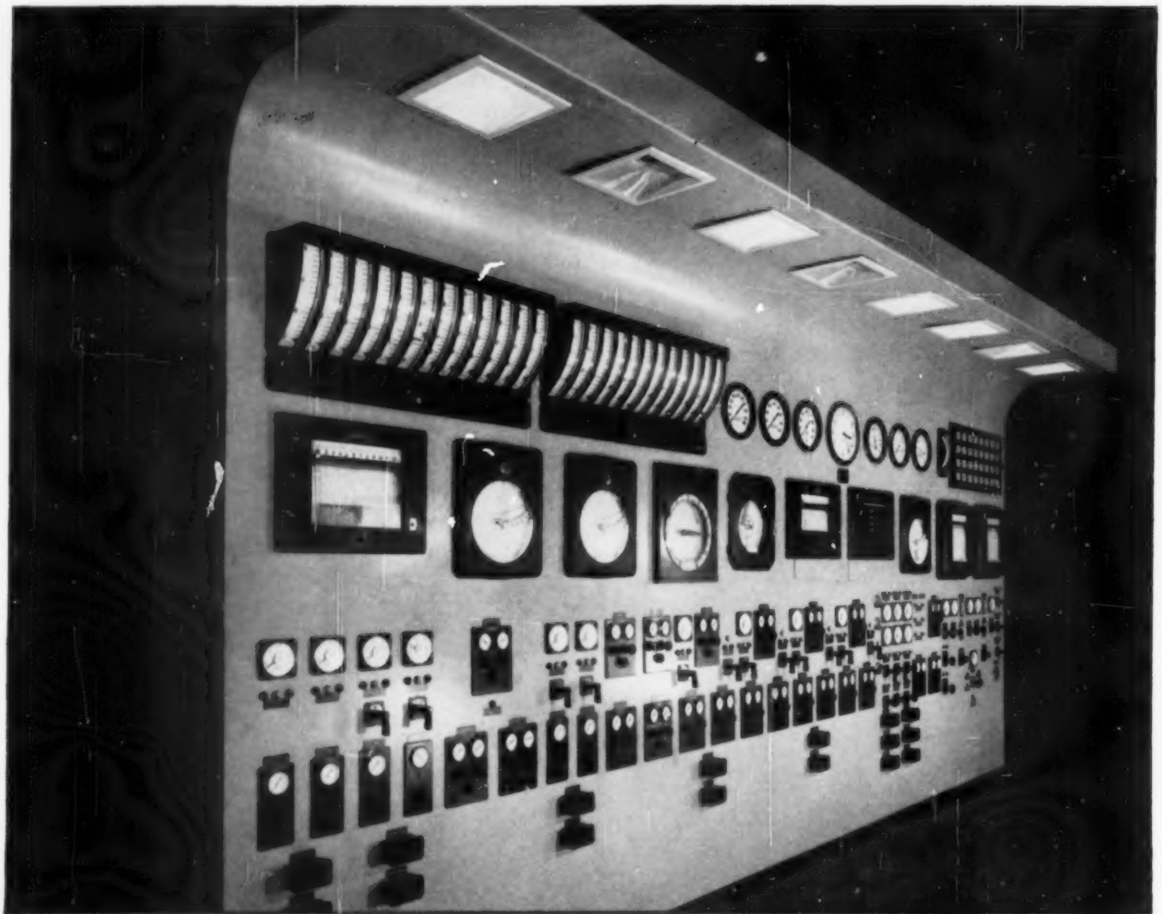
Branch Offices: Clifton, N. J.; Chicago, Ill.; San Francisco, Calif.; Vernon, Calif.; Atlanta, Ga.; Dallas, Tex., Montreal, Canada



**Boiler Operating Panel.** Built by Copes-Vulcan, this panel provides all necessary processing information. It permits firing the furnace or feeding water to the boiler automatically or by remote manual control. Identical Transet controllers and relays simplify maintenance procedures and minimize spare parts stocks.

# Copes-Vulcan Boiler Control installed at Seward Station

Pennsylvania Electric Company's Seward Station uses an integrated Copes-Vulcan system to provide efficient boiler control. Built to maintain a constant main steam-header pressure under all load conditions, this modern control system provides simplicity of circuits and dependable accuracy of components.

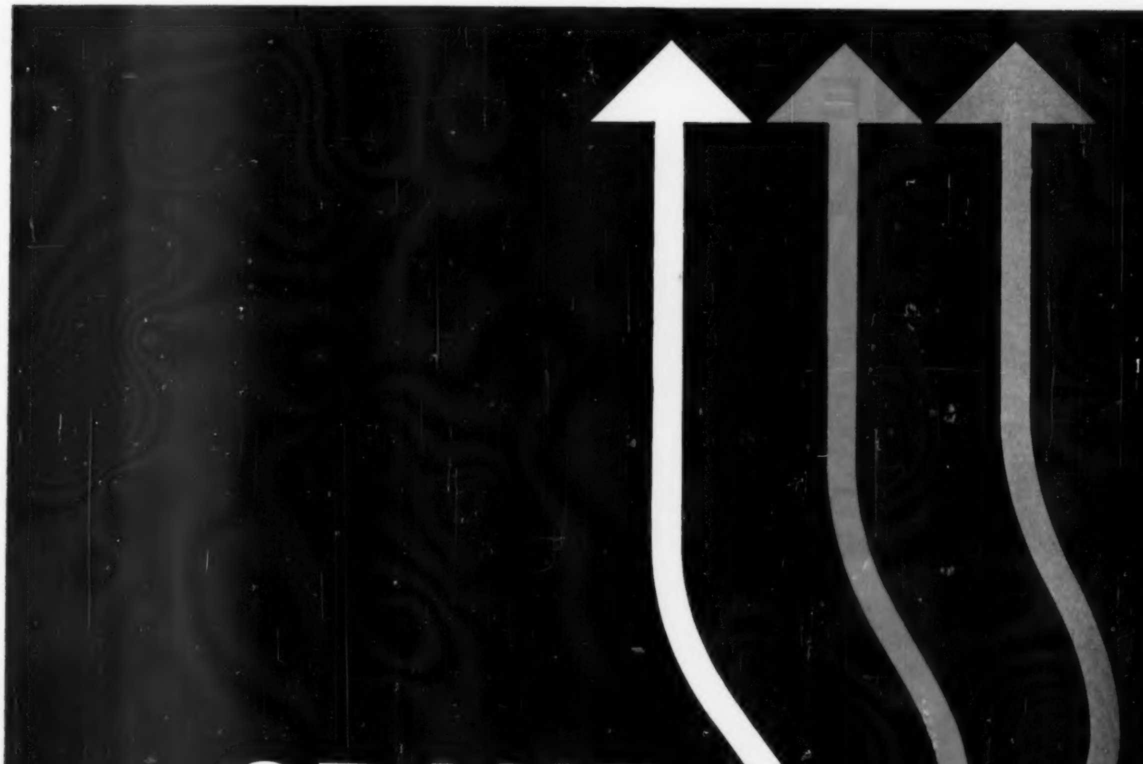
**Correcting pressure changes automatically.** Fuel feed responds to deviations in steam pressure resulting from changes in steam flow rate, and generates the exact amount of steam to restore header pressure. The system also maintains constant drum level.

Mill temperature control, mill-totalizing circuit and boiler feed pump recirculation control are additional functions of this efficient control system.

**Bumpless transfer simplifies operation.** All Copes-Vulcan auto-manual stations provide a continuous picture of performance, permit automatic-to-manual selection without the complications of seal balance.

Noted for modern design, high speed response and precision performance, Copes-Vulcan boiler control systems and Vulcan soot blowing systems cut operating costs at modern stations everywhere.

Available in separate units or integrated into a single package, these control systems are custom engineered to meet individual specifications. For details, write Copes-Vulcan Division, Erie 4, Pa.

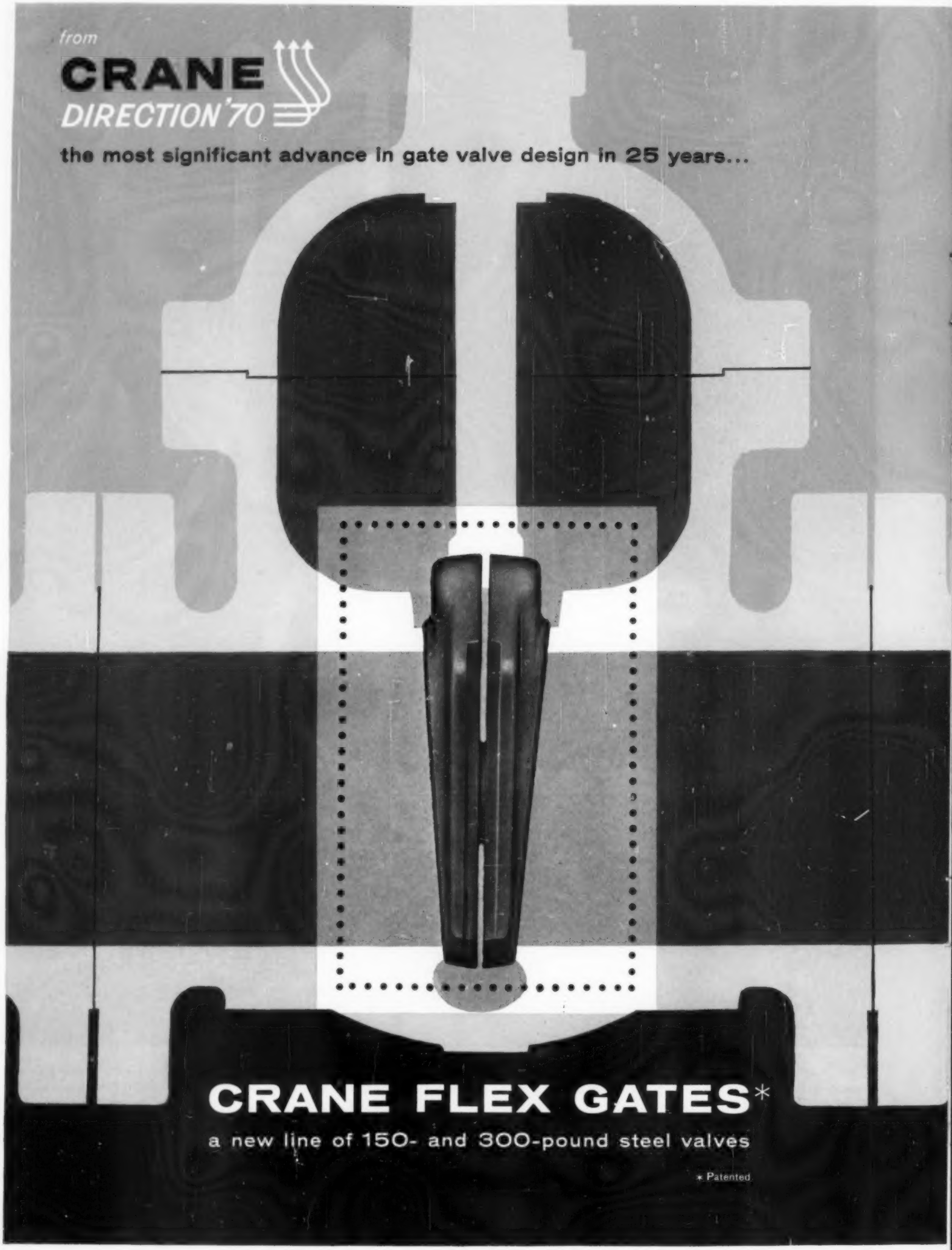


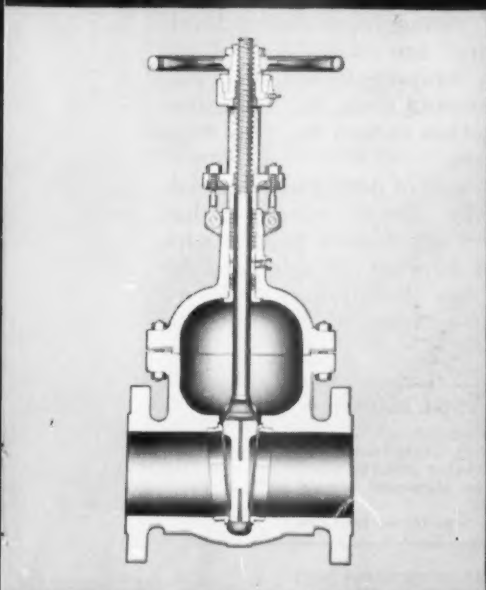

The Soaring Sixties have begun. This is the decade to be marked by accelerated industrial growth. By 1970, predictions are that . . .

- machinery production will double
- petroleum production will increase 4% to 5% each year
- chemicals are to double their present output at the current compound growth rate
- food processing will rise nearly 40%, paper and allied industries will expand about 60%

Total industrial production is estimated to rise some 60% in the next ten years. Looking at it another way, the prospective *increase* in manufacturing and mining is almost as large as the total output of just 12 years ago.

Crane announces Direction '70 . . . new products to help industry meet the challenge of the Soaring Sixties. These are products to improve quality control. These are products to increase production. These are products to help you control your costs. On the following pages is the first . . . the most significant Gate Valve improvement in 25 years. It's the first announcement of many you'll be seeing from Crane in Direction '70.


from
CRANE
DIRECTION '70
the most significant advance in gate valve design in 25 years...
CRANE FLEX GATES*
a new line of 150- and 300-pound steel valves
* Patented



Instead of being made with a solid disc, new Crane Flex Gates have separate disc faces, connected by the spool-like unit you see in the cross section. This joins the two seating faces, yet provides flexibility for the faces to seat tightly with independent action.

# FLEXIBILITY PROVIDES THESE BENEFITS

BECAUSE THEY'RE FLEXIBLE, new Crane Flex Gates seat with less torque.

BECAUSE THEY'RE FLEXIBLE, new Crane Flex Gates unseat with less torque . . . will not stick closed even in high temperature service.

BECAUSE THEY'RE FLEXIBLE, minor deflection of seating faces due to pipe strains does not affect tightness of Crane Flex Gates.

BECAUSE THEY'RE FLEXIBLE, new Crane Flex Gates are tight on inlet seat and outlet seat over a wide range of pressures.

BECAUSE THEY'RE FLEXIBLE, new Crane Flex Gates can be used singly in some services where two conventional gate valves are frequently specified. You can save substantially on piping costs. And they have been exhaustively field tested.

BECAUSE THEY'RE FLEXIBLE, new Crane Flex Gates can be serviced—body seat rings replaced or seating faces refinished—quickly without painstaking accuracy. Slightly off-taper seats do not affect tightness or operating ease.

BECAUSE THEY'RE FLEXIBLE, new Crane Flex Gates will easily outperform any conventional solid wedge disc valve you now use. *And there's no increase in price.*

BECAUSE THEY'RE MADE BY CRANE, these new Flex Gates are completely dependable. You can use them with complete confidence on steam, water, gas, oil or oil vapor service. Stem and disc seating faces are Crane Exelloy. Shoulder-type body seat rings are Exelloy or Crane No. 49 Nickel Alloy. Sizes 12 inch and smaller; 150- and 300-pound pressure classes.

• • • • •

Ask your nearby Crane Distributor for full information on Flex Gates—and for data whenever you work with the products Crane makes. He has the newest in information *and* products. Crane Co., Industrial Products Group, 4100 South Kedzie Avenue, Chicago 32, Illinois.

CRANE®

VALVES • ELECTRONIC CONTROLS • PIPING
PLUMBING • HEATING • AIR CONDITIONING


# HERE'S WHY BEACON COAL

## *WHEN—WHERE—*

| Name of Coal | Character | Mine | Seam | County & State |
|---|---|---|---|---|
| Sonman | Central Pennsylvania Low Volatile | Sonman | "E" | Cambria (Pa.) |
| Colver | | Colver | "B" | Cambria (Pa.) |
| Indian Creek | Central Pennsylvania Medium Volatile | Melcroft | "B" or Miller | Fayette (Pa.) |
| Federal | Northern West Virginia | Federal #1 | Pittsburgh | Marion (W. Va.) |
| Beckley | Southern West Virginia Low Volatile | Eccles #5 | Beckley | Raleigh (W. Va.) |
| Sewell | | Eccles #6 | Sewell | Raleigh (W. Va.) |
| New River | | Stotesbury #8 | Pocahontas #4 | Raleigh (W. Va.) |
| | | Stotesbury #10 | Pocahontas #4 | Raleigh (W. Va.) |
| | | Stotesbury #11 | Pocahontas #4 | Raleigh (W. Va.) |
| Pocahontas | | Keystone | Pocahontas #3 | McDowell (W. Va.) |
| Eagle | Southern West Virginia and Eastern Kentucky High Volatile | Beards Fork | Eagle | Fayette (W. Va.) |
| | | Kopperston #1 | Eagle | Wyoming (W. Va.) |
| No. 2 Gas | | Kopperston #2 | Campbells Creek | Wyoming (W. Va.) |
| Wharton | | Wharton #1 | Hernshaw | Boone (W. Va.) |
| | | Wharton #2 | Hernshaw | Boone (W. Va.) |

As this chart shows, there is a Beacon Coal suited to every fuel need: Domestic or Commercial, Hand or Stoker Firing; General Industrial; Electric Utility; Railroad; Coke or Gas Plants; and Special Uses. Mine locations and shipping facilities permit speedy, efficient deliveries, and conveniently located sales offices mean that there's a Beacon Representative ready, in almost every area, to serve you.

If you would like a copy of this chart for reference—just write or phone the office nearest you.

EASTERN GAS AND FUEL ASSOCIATES
BEACON COAL

**EASTERN**

PITTSBURGH
Koppers Building

# CAN GIVE YOU THE COAL YOU NEED

***AND AS YOU NEED IT!***

| Railroad | District | Sizes Made | Daily Prod. Tons | Name of Coal |
|---|---|---|---|---|
| Penna. | 1 | Egg; Stoker; M/R; N&S; Slack | 2500 | Sonman |
| C & I | 1 | N&S; Slack; Stoker; Nut | 5000 | Colver |
| B & O | 1 | M/R; N&S; Slack | 1200 | Indian Creek |
| Monongahela; B&O | 3 | Lump; Egg; Nut; Stoker; M/R N&S; Slack | 12,500 | Federal |
| C & O; N & W | 7 | Scr. M/R; Stove; N&S; Slack | 2500 | Beckley |
| C & O; N & W | 7 | Scr. M/R; Stove; N&S; Slack | 1000 | Sewell |
| C & O; N & W | 7 | Stoker; M/R; Slack | 2000 | New River |
| C & O; N & W | 7 | Stoker; N&S; Slack | 2500 | |
| N & W | 7 | M/R; Slack | 1800 | |
| N & W | 7 | Egg; Stove; Nut; Stoker; M/R; N&S; Slack | 7500 | Pocahontas |
| N & W | 7 | M/R; N&S | 2900 | Eagle |
| N & W | 8 | M/R; N&S; Slack | 3500 | |
| N & W | 8 | M/R; Nut; N&S; Slack | 7000 | No. 2 Gas |
| C & O | 8 | M/R; N&S | 3500 | Wharton |
| C & O | 8 | M/R; N&S | 4000 | |

# GAS AND FUEL ASSOCIATES

**BOSTON**
250 Stuart Street

**CLEVELAND**
Terminal Tower Building

**DETROIT**
640 Suffield Road
(Birmingham)

**NEW YORK**
60 E. 42nd Street

**NORFOLK**
Citizens Bank Building

**PHILADELPHIA**
Suburban Station Building

**SYRACUSE**
State Tower Building

For New England: NEW ENGLAND COAL & COKE CO.

For Export: CASTNER CURRAN & BULLITT, INC.

HAGAN NEWSLETTER—JUNE

# Behind the panel

## DATA LOGGER ON WHEELS FOR MULTI-PLANT PETROLEUM CORPORATION

Designed for a major European oil producer, this new departure in data processing will consist of a Series 2000 Kybernetes Data Logger mounted in a trailer. The trailer will not only house the 100-point data system, but will also contain its own regulated power supply equipment, so that only the inputs from process transducers and unregulated plant power will have to be connected at the plant site. The Kybernetes system will be used to monitor operating conditions during the various phases of the start-up of new petroleum refining, petrochemical and chemical plants. Later, the equipment will be used for periodic checks throughout the corporation's plant complexes.

In addition to logging 100 process variables, the Kybernetes system continuously scans for off-normal conditions. Process engineers will also use the system for dynamic analysis... selected groups of critical variables will be scanned at the rate of four points per second and recorded on high-speed punched tape for subsequent analysis. Hagan was awarded this order on the basis of a 5-year analysis of available equipment. The corporation's engineers decided that only the Kybernetes system was reliable enough for this type of service. (Details on request--Ask for Item N-1)

## ANOTHER ELECTRONIC COMBUSTION CONTROL SYSTEM FOR STEEL

Another Hagan PowrMag (magnetic amplifier) combustion control system for the steel industry is on order, this time for a bar heating furnace. To be fired with a combination of gas and oil, the furnace will be equipped with Hagan PowrLog recorders and PowrMag controllers. Basic in the selection of a solid state electronic system is the desire to make possible more centralized control, and the ability to transmit operating information over longer distances than is possible with pneumatic systems. The straight-forward engineering, simplicity and high reliability of Hagan PowrMag systems also played a large part in its selection. (Details on request--Ask for Item N-2)

## OPEN HOUSE DAY VISITORS SEE UNSCHEDULED EVENT

A brand new utility central station in the Mid-West was holding open house for their own officials and for invited representatives of other power companies in the area. During the course of the morning, another station in the same network dumped a 100,000 KW load due to electrical difficulties. The resulting surge overloaded the new station and the circuit breakers tripped out. Within fifteen minutes, the station was back in service, having demonstrated to the visitors that all equipment was well able to handle emergencies. Station officials were particularly proud of the performance of the control systems. As shown on the charts, the Hagan combustion, steam temperature and feed water control systems kept operating on full automatic during the unexpected shut-down, start-up period. (Details on request--Ask for Item N-3)

## AUTOMOBILE RADIATORS PROTECTED VIA HAGAN CONTROLS

Automobile radiator thermostats are pre-coated with solder and clamped together before being sent through a muffle furnace for brazing into a complete unit. To maintain flexure life, solder penetration must be firm, yet held to a minimum, therefore accurate control of furnace temperature is a must. The control system uses a Hagan Model P amplifier and a PowrMag controller...with these, furnace temperature of 1400F has been maintained within $\pm 1°F$, well within the required limits. Since solder penetration is a direct function of time and temperature, rejects will be held to a minimum by the accurate regulation made possible by the Hagan controls. (Details on request--Ask for Item N-4)

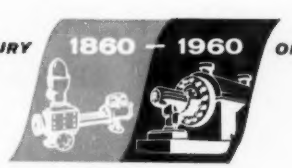
HAGAN CHEMICALS & CONTROLS, INC., Hagan Center, Room 713, Pittsburgh 30, Pa.

**HAGAN** DIVISIONS: CALGON CO. — HALL LABORATORIES — BRUNER CORP.


MACHINES THAT MAKE
ENGINEERING DREAMS COME TRUE

THE PUMP is one of the *oldest* mechanical devices used by man. And "engineers" from pre-history and the middle ages dreamed-up some ingenious machines for the elevation of liquids. Fourteen of these ancient and medieval devices are illustrated and described in this Anniversary Booklet which traces the development of the pump from its earliest beginnings, and also includes a summary of recent Ingersoll-Rand developments in pumping equipment. For your free copy, just send a letter or post card to Ingersoll-Rand, 11 Broadway, New York 4, N.Y., requesting Bulletin 198. There is no obligation, of course.

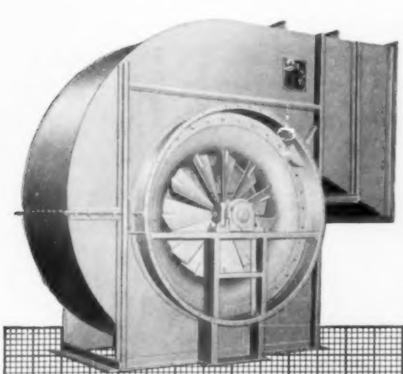
A CENTURY




OF PUMP PROGRESS

Ingersoll-Rand

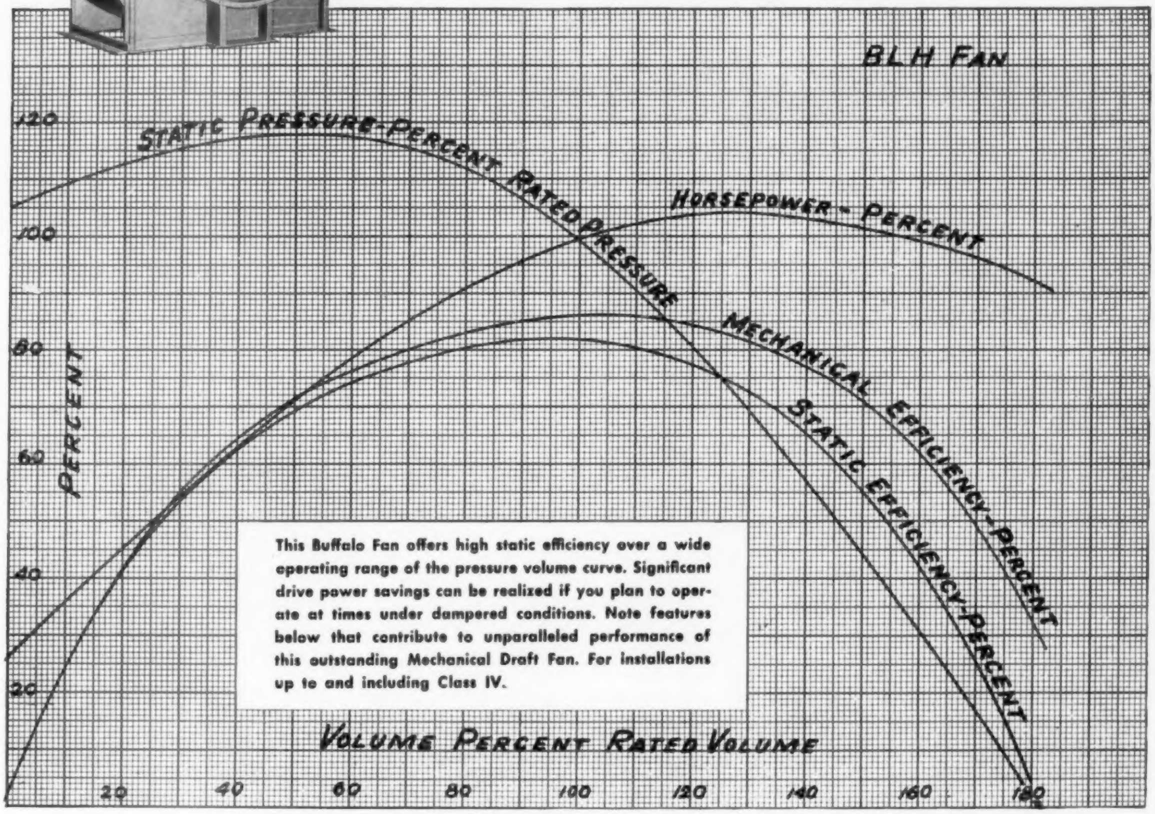
144A10 11 BROADWAY, NEW YORK 4, N.Y.



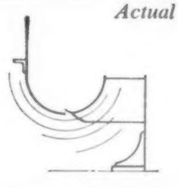
# MAXIMUM EFFICIENCY

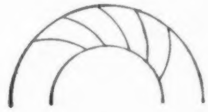
## OVER BROAD OPERATING RANGE

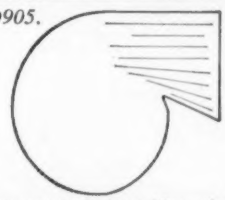
## *THAT'S THE BUFFALO "BLH"*




*Actual performance curve of Type "BLH" Fan. Write for Bulletin FD905.*







**EFFICIENCY AT INLET.** Cross-section of inlet shows smooth passage into wheel formed by curved inlet bell and mating wheel flange. No flat spots to cause turbulence. Fixed or variable inlet vanes available (without reducing efficiency).

**EFFICIENCY IN THE WHEEL.** Showing deep, **backward curve** of blades in "BLH" wheel for smooth, quiet handling of air spun into the wheel in the direction of rotation. No waste turbulence.

**EFFICIENCY THRU THE HOUSING.** Not only is the "BLH" housing streamlined throughout, but its unique divergent outlet delivers air to duct with easy, gradual enlargement for best distribution and static conversion.

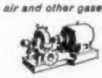
Buffalo

**BUFFALO FORGE COMPANY**

**BUFFALO, NEW YORK**

*Buffalo air handling equipment to move, heat, cool, dehumidify and clean air and other gases.*

*Buffalo Machine Tools to drill, punch, shear, bend, slit, notch and cope for production or plant maintenance.*

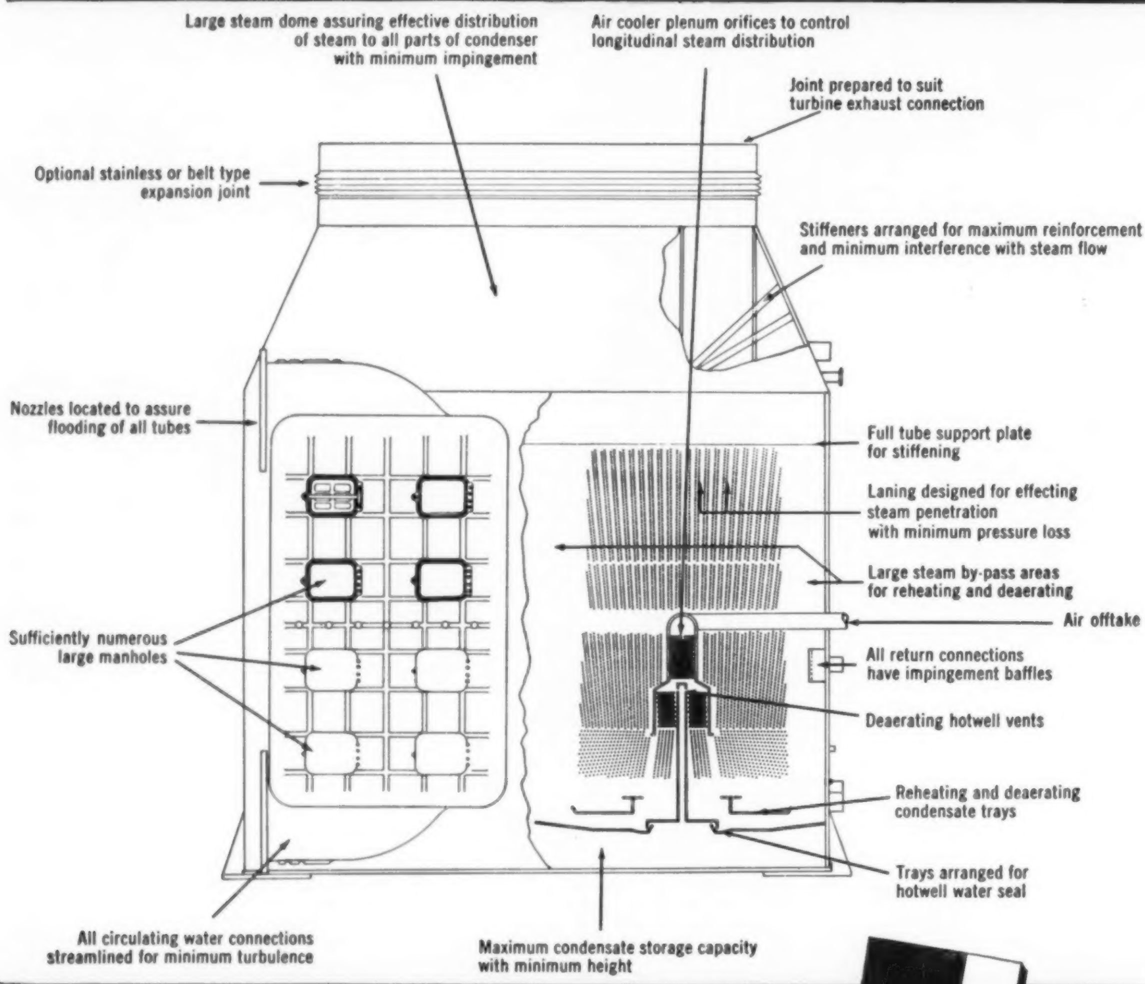


*Buffalo Centrifugal Pumps to handle most liquids and slurries under a variety of conditions.*

*Squier machinery to process sugar cane, coffee and rice. Special processing machinery for chemicals.*

# MARYLAND CONDENSER DESIGN keeps these points in mind—

**• EFFICIENCY • SIMPLICITY • MAINTENANCE • ECONOMY**




**WRITE FOR NEW BROCHURE.** It's yours without cost or obligation.

Industrial Products Division

**Maryland Shipbuilding & Drydock Company**

Baltimore 3, Maryland • *Representatives in Principal Cities*


# DAMPNEY COATINGS LIVE WITH IT!

True measure of a high-heat coating's worth is *continuous operation at rated temperature*. Yet many so-called "heat-resistant" coatings take only occasional peaks — fail rapidly in 'round-the-clock service.

Dampney coatings are rated always for day in, day out operation at maximum temperatures. Hold them to it, if schedules call for steady heat, or let them fluctuate to ambient and back. Either way, Dampney silicones and ceramics give you full protection — with plenty in reserve.

Most important, Dampney coatings are selected to meet specific conditions of operation, temperature and corrosive environment. Thus they establish a lasting foundation easily maintained and permanently ending time-consuming and costly surface preparation.

Repeat orders — from a typical customer, 26 in 12 months for enough material to protect 1,929,000 square feet of steel — is the best evidence we have that when industry wants honest high-temperature coatings, it remembers Dampney silicones and ceramics, identified by the two trade names, DAMPNEY and THUR-MA-LOX.

We suggest you do likewise when you want real protection — resistant to 1000°F., to atmospheric corrosion, and to weather exposure — for these industrial hot spots . . .

| | |
|---|---|
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| **forced and induced draft fans** | **incinerators** |
| **heat-treating furnaces** | **pulverizers** |
| **autoclaves and retorts** | **blast and open hearth furnaces** |

Remember, too, the first Dampney trade name and product, known and used today the world around, APEXIOR NUMBER 1 for boiler interiors. For all hot metal, wet or dry, the best protection available is made and marketed by

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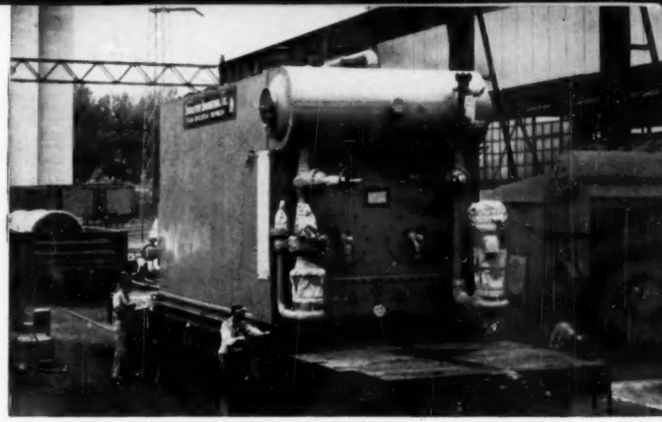

**To 90,000 lb/hr . . . Type VP**—A compact, economical, natural circulation, *package* boiler for the majority of industrial requirements. Steam capacities from 4,000 to 90,000 lb/hr; pressures to 700 psi; temperatures to 750 F. Most sizes shipped complete with firing equipment for oil or gas, setting and forced draft fan arrangement.

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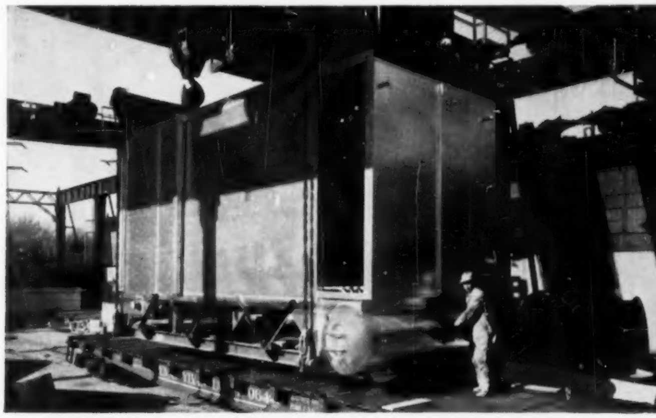

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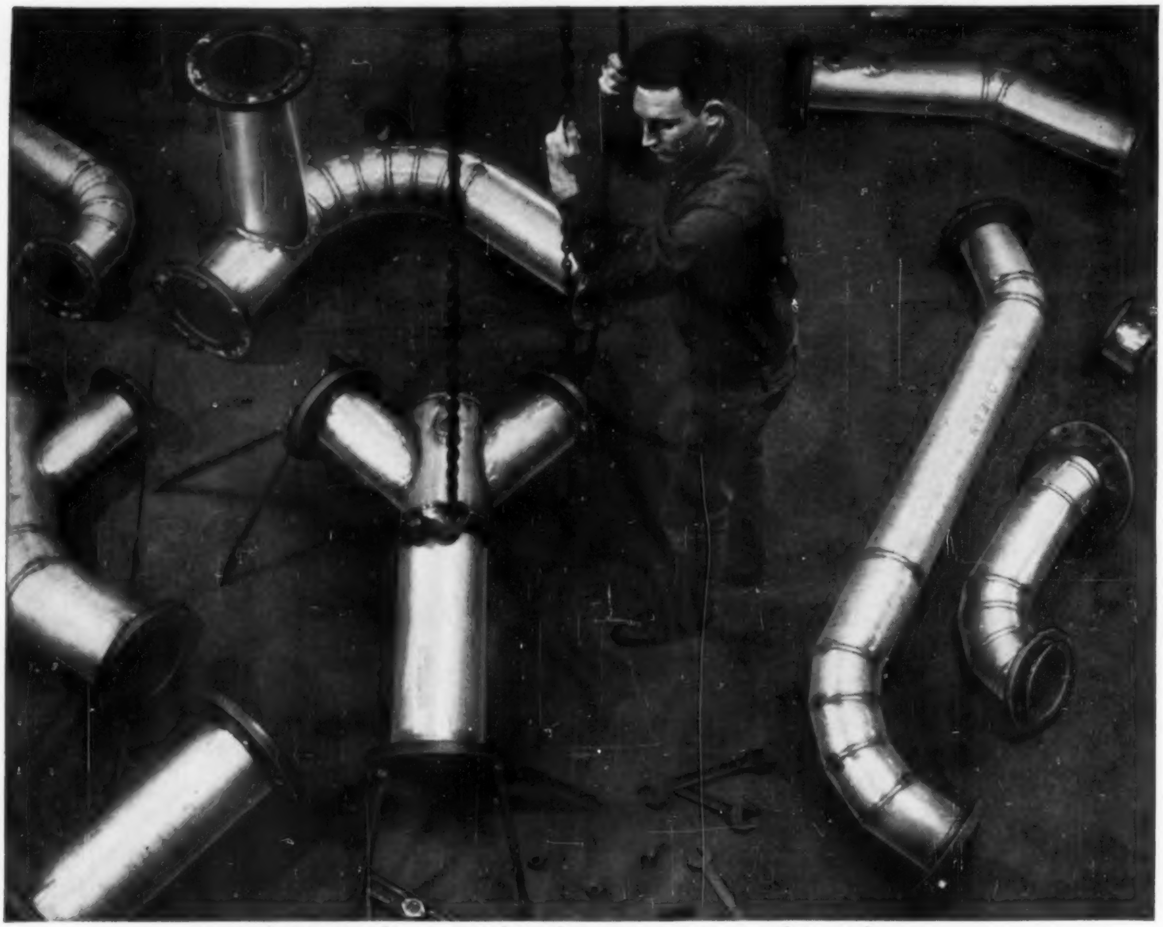

**CUPRO NICKEL** salt water circulating line sections to serve auxiliary condensers aboard gypsum rock carriers recently fabricated by Boro Marine & Industrial Corp.

# Wide variety of Cupro Nickel piping for salt water lines is readily made from sheet by welding

Circumferential joints are welded manually. Longitudinal seams are welded in mechanized equipment by inert-gas, metal-arc processes.

Ever higher velocities in salt water lines and the growing economic importance of continuity in service, on shipboard, and also in tidewater power plants and oil refineries, are leading to increasing use of Cupro Nickel in piping.

Techniques and skills for economical fabrication of even the most complicated elements of Cupro Nickel piping systems are keeping pace. Boro Marine & Industrial Corp., Port Richmond, Staten Island, N. Y., a specialist in the field, forms piping in sizes 6" to 24" diameter from Anaconda Cupro Nickel stock sheet, usually 48" x 96" x 3/16". Elements shown above indicate the variety possible. Seamless tubing is used for smaller diameters.

Boro Marine fabricates piping from both Cupro Nickel 30%-702 and Cupro Nickel 10%-755. The trend, however, is to Cupro Nickel 10%-755 for the majority of salt water line installations on commercial vessels and in industrial jobs, according to M. E. Wuensch, president. The alloy was developed by Anaconda for this kind of service. It is resistant to corrosion by both clean and polluted sea water, even at relatively high velocity of flow, and is resistant to corrosion by sea water containing air bubbles.

**TECHNICAL ASSISTANCE.** For help in selecting the alloy best suited for a particular job in heat transfer and piping systems, call in your American Brass representative, or write: The American Brass Company, Waterbury 20, Conn. In Canada: Anaconda American Brass Ltd., New Toronto, Ont. 6068

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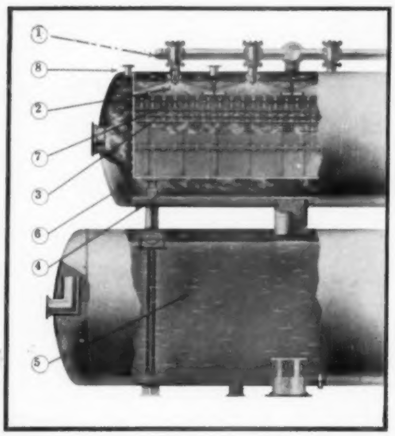
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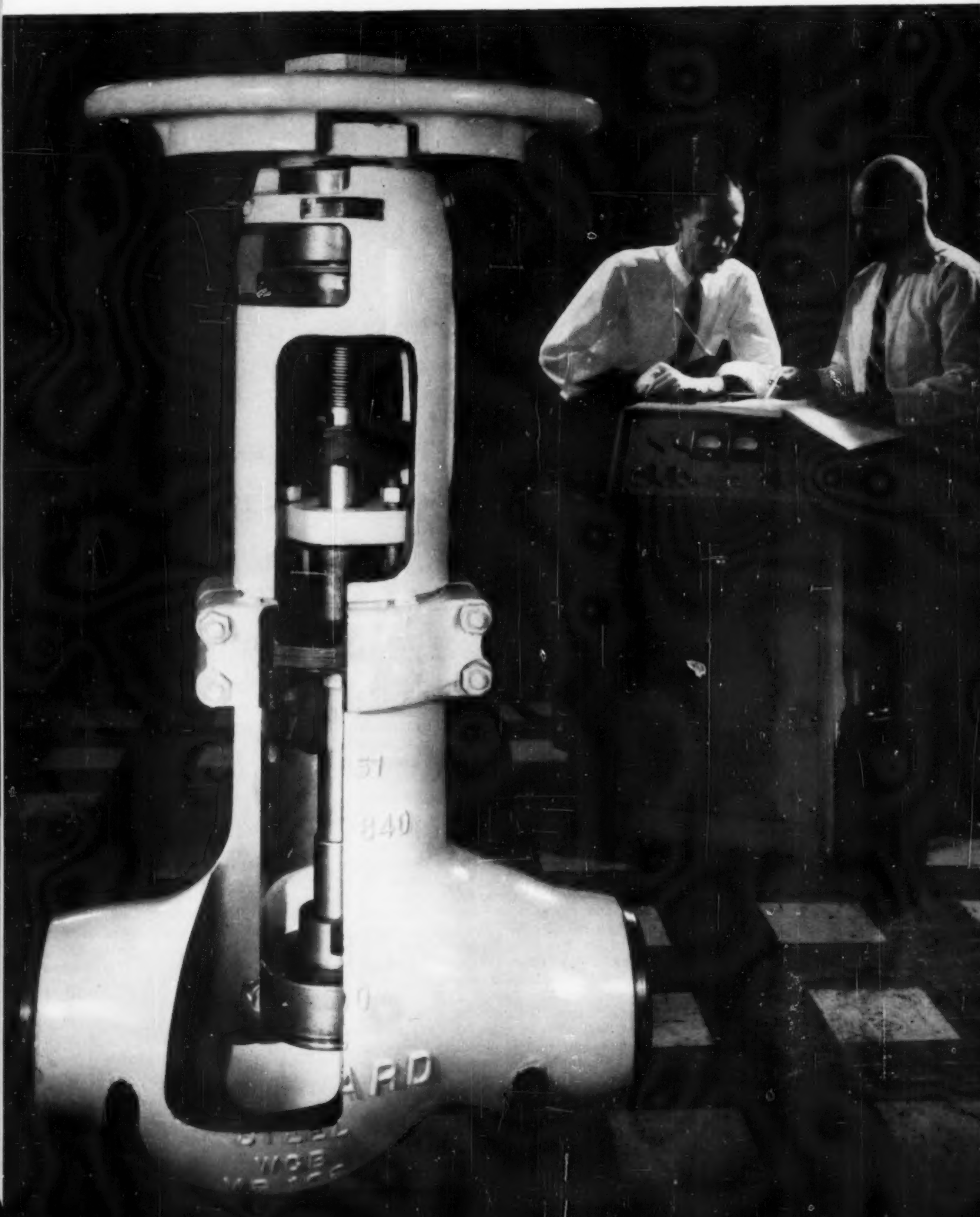
# The Story of Edward Research



# and the Pressure-Seal* Valve

The introduction, in 1945, of commercial steel valves with pressure-seal bonnet joint construction was a significant advance in the art of building large high-pressure steel valves. The new pressure-seal construction eliminated massive flanged bonnet joint connections and made it possible to seal the valves permanently at high temperature. These advantages were immediately apparent to users everywhere. But bonnet joint leakage problems developed on the new valves in service. And Edward engineers soon became convinced that the original 45° pressure-seal gasket design used by Edward and others required improvement. Here's the Edward pressure-seal redesign story:

**INITIAL TESTING**—Edward engineers began their studies on leak problems in 45° gasketed pressure-seal valves by determining under what conditions the valves were most likely to leak. It was discovered that when sealing areas were thoroughly degreased and tested with air, the pressure seal would leak at all pressures above 10 psi. In addition, minute imperfections in the sealing area of the valve body bore were preventing intimate gasket contact and contributing to leakage. Edward engineers found that a broader area of contact between gasket and bonnet and between gasket and body bore would also reduce leakage. Further investigation showed that a gasket plating material softer than the silver plating previously used would be helpful in achieving better contact in the sealing area.

To obtain more gasket sealing area, experiments were conducted on gaskets with angles from 45° down to 17°. Over a period of several months, tests were made with 25 different combinations of gasket and bonnet angles and various gasket plating materials. Assembled valves were tested for tightness with air pressures ranging from 10 psi to 2160 psi, and at temperatures up to 1000 F for extended periods. And tests on ease of valve disassembly (an important user consideration) were made with each gasket combination.

**TEST CONCLUSIONS**—At the end of their extensive testing program on the pressure-seal design, Edward engineers were able to draw these conclusions:

1. 45° gaskets, which showed no leak on water tests, leaked air at similar pressures.
2. No gasket of any design, either plain or with a plating of 100 Brinell or harder, would seal air when assembled unlubricated.
3. The best sealing under test resulted with a 25° gasket angle which was 1° more acute than the bonnet angle.
4. Gaskets of 25° angle actually tripled the sealing surface area.
5. A stainless inlay in the valve body bore sealing area substantially improved gasket-to-body contact and provided a corrosion-resistant surface.
6. Gaskets of 25° angle, plated properly with a malleable coating, gave perfect air or steam sealing at all temperatures, whether assembled dry or with a lubricant.

**NEW VALVE DESIGN**—As the result of their tests, Edward engineers designed the completely new pressure-seal valve with 25°-65° joint (see diagram below). This change in gasket angle, with an increase in sealing surface area, and the addition of a special corrosion-resistant malleable coating, brought an end to bonnet joint leakage in Edward pressure-seal valves. And, since 1953, thousands of Edward pressure-seal valves have been installed in a great variety of services *without a single case of failure reported to date.*

The story of Edward research and the pressure-seal valve is typical of the kind of research progress and product leadership you can expect from Edward Valves. Edward builds a complete line of forged and cast steel valves from ⅛" to 18" for industrial, marine, petroleum and technological services. For more detailed information, contact your Edward Representative, or write Edward Valves, Inc., 1206 West 145th Street, East Chicago, Indiana. Subsidiary of Rockwell Manufacturing Company. Represented in Canada by Lytle Engineering Specialties, Ltd., 438 St. Peter Street, Montreal.

*Patented

**EDWARD STEEL VALVES**

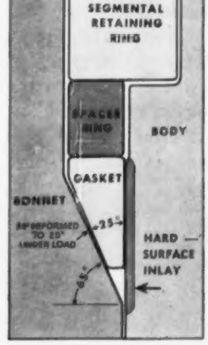
another fine product by

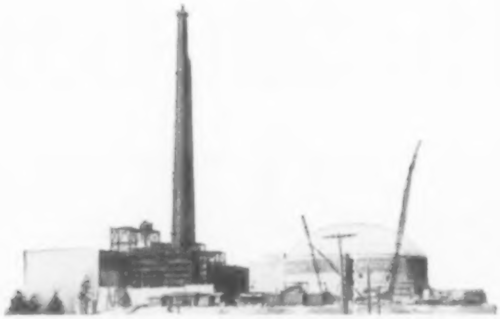
**ROCKWELL**

**P. J. DUKES**, product development engineer, and D. MacGregor, works manager, are shown with helium leak detector used in testing pressure-seal valve joints.

**SPECIAL GAS-FIRED FURNACES** were used in temperature testing of new pressure-seal valve design. Furnaces heated valves up to 900 F for 24 hour periods. Pressure was maintained with hydro-pneumatic pumps.

NEW 25° GASKET DESIGN





# INDIAN POINT

*Indian Point Station as it looked in April this year, showing some of the piping to be installed by Kellogg's Power Piping Division*

# NUCLEAR POWER PIPING BY KELLOGG

Construction progress at Consolidated Edison Company's Indian Point Station demonstrates how Kellogg's broad erection experience can take tomorrow's newest and toughest power piping requirements in stride.

At this unique 275 Mw nuclear steam electric generating station, Kellogg has a contract to manufacture, deliver, and to erect all stainless and carbon steel nuclear piping for the inside of the reactor sphere, and all power piping for the conventional portion of this plant. Kellogg also stress-analyzed the major portion of this piping. Much of the stainless piping will be manufactured in Kellogg's Williamsport plant.

The particularly rigid specifications of high quality and close tolerances required the assignment of a special engineering staff to the site. This staff plans, coordinates and supervises each step of Kellogg's erection assignment. One important phase entails over 2200 critical welds, most utilizing Kellogg's K-Weld technique. Another is the radiographic inspection of each weld, which Kellogg is undertaking with its own equipment and personnel.

---

Kellogg welcomes inquiries for its stress analysis, metallurgical, engineering, manufacturing and erection services.

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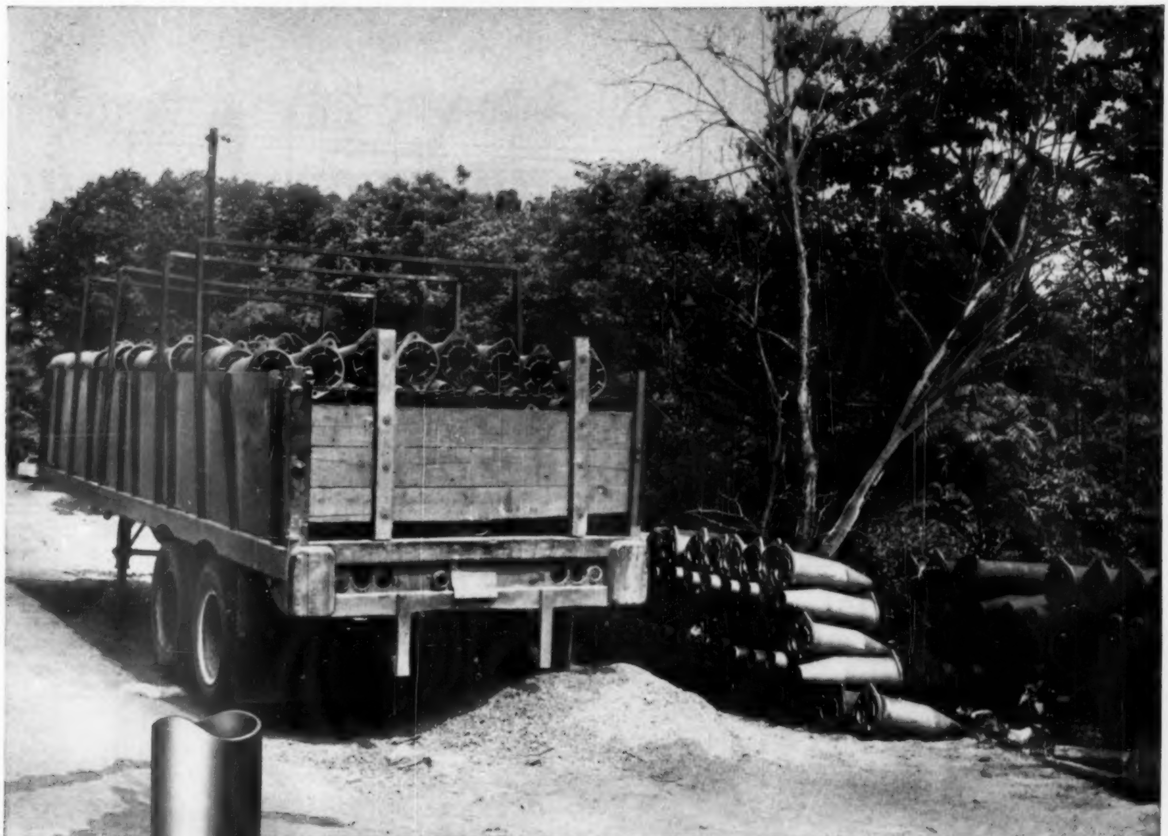

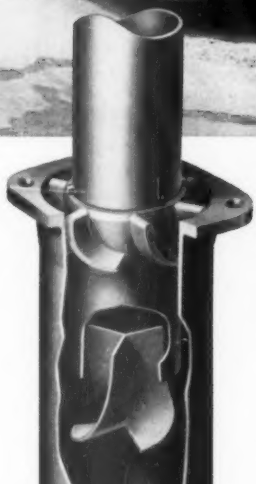

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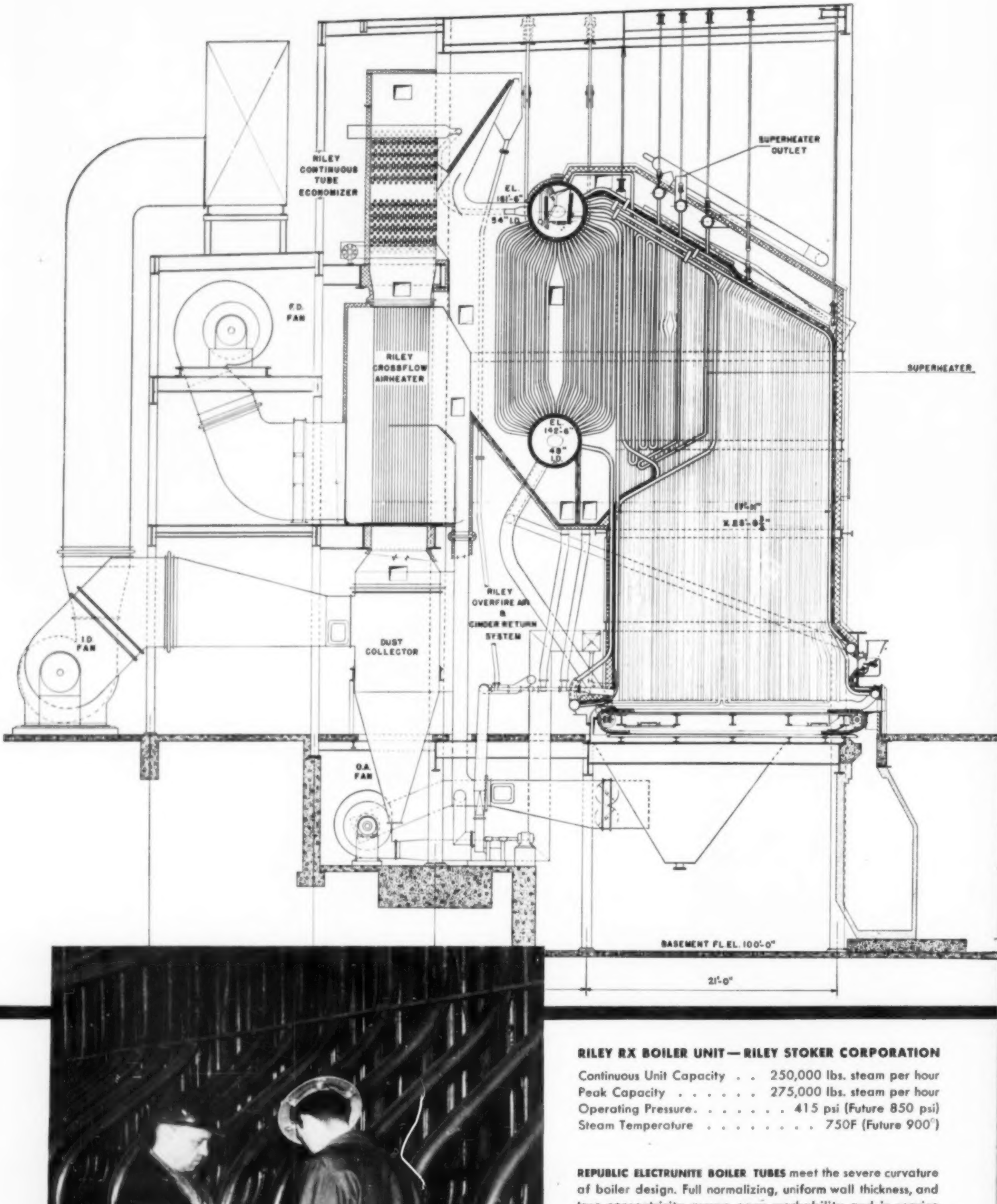


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| Operating Pressure. . . . . . . | 415 psi (Future 850 psi) |
| Steam Temperature . . . . . . . . | 750F (Future 900°) |

**REPUBLIC ELECTRUNITE BOILER TUBES** meet the severe curvature of boiler design. Full normalizing, uniform wall thickness, and true concentricity assure easy workability and in service dependability. Bends are smooth and uniform. Rolling-in operations are easier. Results: greater savings in installation time and operating costs.

# *more power for the city of Painesville...with*

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Push a button, flip a switch—power to operate schools, factories, and the convenience of modern living originates in the boiler units of power plants. And a great share of boilers are constructed of Republic ELECTRUNITE® Boiler Tubes.

Painesville, Ohio, a growing community in the heart of industrial America, has just installed an additional Riley RX Boiler Unit in its municipal power plant.

Designed, engineered, erected by Riley Stoker Corporation, Worcester, Massachusetts, the RX has a continuous unit capacity of 250,000 lbs. of steam per hour, peak capacity of 275,000 lbs. of steam per hour and an operating pressure of 415 psi at a steam temperature of 750F. The RX Unit is fired by a Riley Traveling Grate Spreader Stoker.

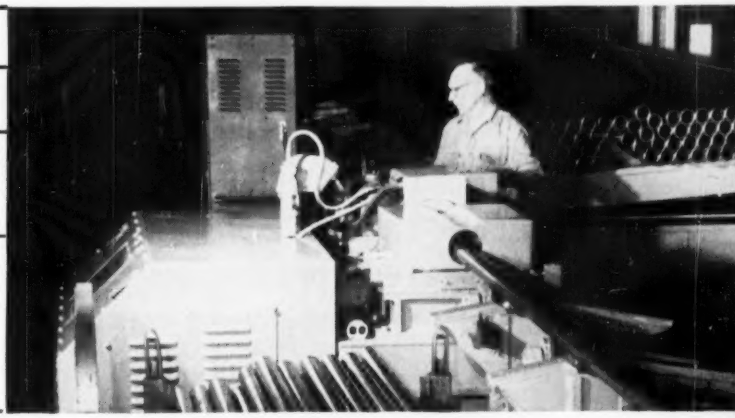
Republic ELECTRUNITE Boiler Tubes are accepted by boiler designers, engineers, and manufacturers. ELECTRUNITE is produced to meet ASTM specifications, the ASME Boiler and Pressure Code, and boiler insurance requirements.

For boiler, heat exchanger, condenser, or evaporator tubes, get ELECTRUNITE facts first-hand. Call your Republic representative, or write direct.

**FARROWTEST REJECT TABLE**

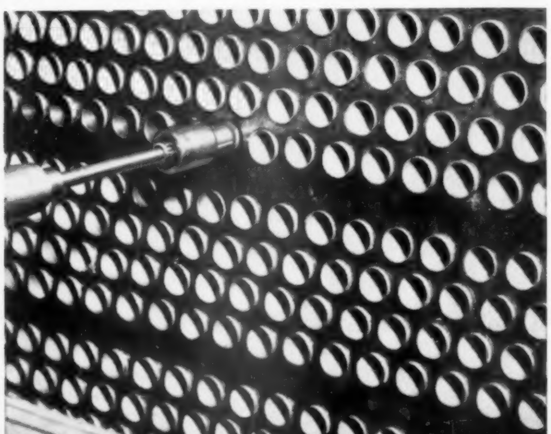
| Wall Thickness (B.W. Gage) | Minor dimension of the defect (Length or Depth) | Defective Area (Length, Depth Plane) |
|---|---|---|
| 20 | .006" | .0025 sq. inches |
| 18 | .006" | .003 sq. inches |
| 16 | 12½% of wall | .003 sq. inches |
| 14 and 13 | 12½% of wall | .004 sq. inches |
| 12 and heavier | 12½% of wall | .005 sq. inches |

**FARROWTEST** detects and rejects not only tubing containing defects which completely penetrate the wall; but also tubing with defects equal to, or greater than, those shown in this table. For irregular defect shapes, a tube with defect area equal to or greater than shown above is rejectable. Where required, sensitivity of FARROWTEST equipment can be calibrated to reject defects of lesser specified area than shown in table, at extra cost.



**FARROWTEST—QUALITY YOU CAN MEASURE.** Not a laboratory theory, not a mere inspection tool, but an exclusive production test that detects and rejects tubing containing defects of critical size. FARROWTEST is offered as an alternative to other less positive tests in accordance with the table above, at no extra cost.

**REPUBLIC ELECTRUNITE STAINLESS STEEL HEAT EXCHANGER TUBING** resists corrosion and provides long trouble-free service for main surface condensers in utility power plants where water is used directly from natural sources. Call your Republic representative for additional information, or send coupon.



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Combustion

## EDITORIAL

# On Mountain Climbing

How refreshing it is to be wrong, and know it! The plight of the climbing party recently stranded on an Alaskan mountain top had us very much concerned for a time. Why, we thought, should these foolhardy adventurers having got themselves into trouble, ask hundreds of rescuers to leave their homes at risk of life itself to save them?

And then came the light. It was not our place to judge. The rescuers went to the task of their own free will—seeing only that fellow beings were in peril and needed aid. This was a sample of humanity at its best and the matter was, in the words of Lincoln, "far above our poor power to add or detract."

Of the climbers themselves what can be said? Thank God we have people such as these among us. So long as there are in our society men who must climb a mountain simply because it is there we have little to fear from social decay. So long as we can find men to tackle a challenge simply because it is challenging we shall continue to forge ahead in all the arts and sciences.

In engineering as in the exploration of frontiers and mountains these are the men who matter—who have always mattered. These are the men who will be rewarded, by dollars today or renown tomorrow. I doubt that either means very much to them.

Here are the engineers who are privileged to give to the world because something in their nature demands that they go beyond the world's boundaries of thought and knowledge. Out beyond the confining walls of test data and human know-how they venture because they must. Out there they will seek and discover and decide. And these are the ones we shall call "great."

The May issue of COMBUSTION carried the first part of this paper. That part discussed the major considerations behind an economic evaluation of a condenser and the study results of 650 separate condensers the digital computer was asked to solve. This part gives the actual outline of data and formula submitted for programming on the digital computer.

# Economic Sizing of Condensers Through the Use of the Digital Compute—II*

By

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**AMI LEIDNER**[§]
Commonwealth Edison Company, Chicago, Illinois

**CONRAD E. PERSON**[‖]
Commonwealth Edison Company, Chicago, Illinois

## THE PROBLEM

To determine the economic sizing of a condenser for a 300-Mw unit the following variables were used:

1. Nine condenser surfaces, ranging from 150,000 to 230,000 sq ft at 10,000-sq ft increments.
2. Eight tube lengths varying from 38 to 52 ft in increments of 2 ft.
3. Three tube diameters: $^{6}/_{8}$, $^{7}/_{8}$ and 1 in.
4. Three cooling water velocities: $6^{1}/_{2}$, 7 and $7^{1}/_{2}$ fps.

To determine the optimum condenser size the total present valued cost of each size was calculated, covering a 35-yr period. This included the carrying charge on investment, coal, and pumping cost, and capability evaluation.

The investment cost comprised the cost of the shell, tubes, erection, erection supervision, circulating water pumps and motors, spring supports and support jacks, circulating water pipes, valves and traveling screens.

Items which cost the same for all condenser sizes, such as steam jet air evactors and hotwell pumps, were not included in the investment costs.

The pumping cost covered the cost of the energy required to overcome the total resistance of the circulating water system, including the intake and discharge tunnels, screens, condenser, piping and valves.

* Presented at the American Power Conference, Chicago, Ill., March 29–31, under the same title and published by permission of the Conference.
† General Mechanical Engineer.
‡ Engineer, Office of Manager of Engineering.
§ Engineer, Mechanical and Building Engineering Department.
‖ Methods Analyst, Program Division Revenue Accounting Department.

The cost of fuel was computed by using four loads, namely: 325, 284, 203 and 144 Mw; four circulating water inlet temperatures: 38.87, 45.62, 54.5, and 65.92 F.

For one condenser size and one circulating water inlet temperature, the heat transfer rate "U," from steam to cooling water, is determined for a particular load and at an assumed 1-in. Hg vacuum according to the Heat Exchange Institute Standards. The weight of steam in lb per hr entering the condenser and the heat rejected per lb of steam to the cooling water are determined. Using these factors the computer calculates the vacuum. If a difference between assumed 1-in. Hg. condenser pressure and the calculated pressure is 0.01 in. Hg or less, the calculated pressure is accepted for the particular load and water temperature. If this difference is greater than 0.01 in. Hg the calculated pressure becomes, in turn, the assumed pressure and the process is repeated until the accepted pressure is obtained. For every pressure, at a particular load, the heat rate is found from the correction curves. The heat rate multiplied by the total number of kwhr and by the cost per Btu of coal results in the cost of coal for a particular load and circulating water inlet temperature.

The circulating water pumping cost is calculated separatelv for two cases:

1. Both pumps operating at all loads and circulating water inlet temperatures.
2. Both pumps operating as above except for the two upper loads and two lower circulating water inlet temperatures when only one pump operates.

For each vacuum the turbine will have a different capability evaluation. The 315-Mw load at $3^{1}/_{2}$ in. Hg

is used as the capability base and is subtracted from the number of kilowatts corresponding to the computed vacuum for each alternate. This difference, multiplied by the investment cost per kilowatt, determines the value of the additional output of the turbine with respect to the above base. Since it was considered that maximum demand on the system would occur when the circulating water temperature was at 70 F, capability differences were evaluated only for this temperature. Realizing that due to the overhaul schedule of other generating capacity the circulating water might be a limitation on capability at other times of the year, the capability evaluation was made at both 90 and 100 per cent of the summer peak capability. The two percentages gave the differences between the two capability charges.

The power used by the circulating water pumps was subtracted from the capability of the machine in determining the capability evaluation.

## TO DETERMINE

I. Investment Cost, Present Valued
II. Pumping Cost, Present Valued, Two-Pump Operation
IIa. Pumping Cost, Present Valued, One- and Two-Pump Operation
III. Coal Cost, Present Valued, Two-Pump Operation
IIIa. Coal Cost, Present Valued, One- and Two-Pump Operation
IV. Capability Evaluation, Present Valued
V. Total Cost, Present Valued, Two-Pump Operation *Only:* I + II + III − IV = V
Va. Total Cost, Present Valued
For two-pump operation at all loads and temperatures, except for one-pump operation for the following: 325 Mw at 38.87 and 45.62 F. 284 Mw at 38.87 and 45.62 F. It was determined that the unit would probably run with one circulating pump at these two loads and at all temperatures up to and including 45.62 F.

## GIVEN VALUES

Condenser surface ($S$) 150,000, 160,000, 170,000, . . . . . 230,000 sq ft
Cooling water velocity ($V$) 6.5, 7.0 and 7.5 fps
Actual tube lengths 38, 40, 42, 44, . . . . ., 52 ft
Effective tube lengths ($L$) 37.75, 39.75, 41.75, 43.75, . . . ., 51.75 ft
Diameter of tube 0.750, 0.875 and 1.000 in.

I. Investment Costs, Present Valued = (A + B + C + D + E + F + G + H) × Factor "X"

The factor represents the present value of the carrying charge for the 35-yr life of the equipment. It includes insurance, state and local taxes, depreciation, interest, equity return and federal income tax. Since most of the above items vary from year to year and from one plant location to another, it is necessary to check this factor each time a condenser or unit is calculated.

A. *Shell Cost*

| Cost Per Sq Ft | Length of Tube, Ft |
|---|---|
| $3.06 × 1.15 | 38 |
| 3.01 × 1.15 | 40 |
| 2.96 × 1.15 | 42 |
| 2.90 × 1.15 | 44 |
| 2.84 × 1.15 | 46 |
| 2.78 × 1.15 | 48 |
| 2.73 × 1.15 | 50 |
| 2.69 × 1.15 | 52 |

The factor 1.15 reflects the increased book price of the condenser shell since the publication of these figures up to the date of this condenser study.

| Diameter of Tube, In. | Diameter Factor |
|---|---|
| 1.000 | 1.00 |
| 0.875 | 0.95 |
| 0.750 | 0.90 |

Shell cost = condenser surface × cost per sq ft × diameter factor

B. *Erection Cost*

Condenser surface ($S$) × $0.85 × 1.15 = erection cost

C. *Erection Supervision Cost*

Condenser surface ($S$) × $0.13 × 1.15 = erection supervision cost

D. *Cost of Tubes*

| Diameter, In. | Sq Ft Per Lineal Ft | Lb Per Linear Ft | Base Price, $ Per Lb* |
|---|---|---|---|
| 0.750 | 0.1963 | 0.419 | 0.6812† |
| 0.875 | 0.2291 | 0.493 | 0.6754 |
| 1.000 | 0.2618 | 0.568 | 0.6633 |

* Over 30 ft tubes up to and including 40 ft add $0.08 per lb. Over 40-ft tubes up to and including 50 ft add $0.10 per lb. Over 50-ft tubes up to and including 60 ft add $0.12 per lb.
†Published prices:
Sq ft per linear ft × effective length = area per tube, sq ft
Condenser surface ÷ area per tube = number of tubes
Number of tubes × actual length of one tube = total tube length-ft
Total tube length × 1.05 lb per ft = total weight, lb‡
Base price + extra price (if above 30 ft) = price of tubing, $ per lb
Price of tubing × total weight = cost of tubes
‡ Weights: The actual weight per foot of condenser tubes is higher than the published theoretical weight for commercial tubes of the same alloy. Five per cent has been added to the published weight to account for this increase in weight.

E. *Circulating Pump Cost*

Cost of the pump is determined for each case according to the following formula:

Cost of pump = 0.5533 × gpm (1 + factor)

The 0.5533 is based on a book value $503 per 1000 gpm + 10 per cent for price increase.

The *factor* in the formula represents the deviation from the standard head as follows:

| Head, ft | Factor |
|---|---|
| 15–25 | 0.075 + 0.04 × (25 − required head) |
| 26–40 | 0.005 × (40 − required head) |
| 41–55 | 0.005 × (required head −40) |
| 56 and up | 0.075 + 0.04 × (required head −55) |

Total pumping head = friction head in tubes + water box resistance + circulating water pipe, valves and expansion joint resistance + traveling screens resistance + tunnel resistance.

Tube and water box resistance per Heat Exchange Institute Standards are as follows:

FRICTION HEAD IN TUBE
(Ft of water per linear ft)

| Velocity, Fps | 1* | 2 | 3 |
|---|---|---|---|
| 6.5 | 0.350 | 0.283 | 0.234 |
| 7.0 | 0.400 | 0.325 | 0.269 |
| 7.5 | 0.454 | 0.370 | 0.304 |

*Tube OD, In.—[1]0.750 [2]0.875 [3]0.304

WATER BOX RESISTANCE
(Ft of water)

| Velocity, Fps | Resistance |
|---|---|
| 6.5 | 1.265 |
| 7.0 | 1.418 |
| 7.5 | 1.570 |

The circulating water pipe is assumed to be sized for a velocity of about 10 fps resulting in an average head loss of approximately 0.6 ft per 100 linear ft for all flows. The average head loss in fittings, valves and expansion joints is found to be approximately 2.4 ft for all flows. The circulating water system head loss, excluding the condenser tubes and water box is therefore 2.4 + 0.6 = 3 ft, based on an expected 100 ft of pipe.

The tunnel and screens resistance is formulated as follows:

$$\text{Tunnel and screens resistance} = 2.5 \times 10^{-11} \times (\text{gpm})^2, \text{ ft}$$

The above equation was developed by the company hydraulic engineer.

The circulating water flow is calculated as follows:

$$\text{Circulating water flow} = \frac{\text{condenser surface} \times \text{velocity}}{\text{effective length} \times K}$$

$K$ is a constant depending on the tube diameter, length and gage, and number of passes.

FOR SINGLE-PASS CONDENSER, 18 Bwg

| Tube—OD, In. | $K$ |
|---|---|
| 0.750 | 0.188 |
| 0.875 | 0.155 |
| 1.000 | 0.131 |

F. *Circulating Pump Motors Cost*

$$\text{Hp} = \frac{\text{total resistance} \times \text{gpm} \times 8.3368}{33{,}000 \times 0.906 \times 0.85}$$

8.3368 = lb per gal at 60 F: 0.906 = motor efficiency; 0.85 = pump efficiency.

$$\text{Hp} = \frac{\text{total resistance} \times \text{gpm} \times 8.3368}{25{,}413.3}$$

$$\text{Cost of circulating pump motors} = \$50{,}000 \times \text{hp}$$

G. *Springs and Support Jacks*

Cost of jacks = \$.078 × 1.15 × condenser surface
= \$.08970 × condenser surface
Cost of bolts = \$.031 × 1.15 × condenser surface

These costs were not included in this calculation since this condenser was mounted solidly upon concrete and the expansion was taken care of by a "dog bone" expansion joint.

H. *Cost of Pipes and Valves*

| Circulating Flow Range, Gpm | Cost of Pipes, Valves, Fittings and Screens |
|---|---|
| 150,000–150,900 | \$ 88,564 |
| 151,000–179,000 | 113,500 |
| 180,000–199,000 | 114,500 |
| 200,000–209,000 | 128,000 |
| 210,000–299,000 | 130,000 |

II. Pumping Cost, Present Valued, Two-Pump Operation

Pumping cost, period I =

$$\text{hp} \times \underset{(1)}{0.746} \times \underset{(2)}{\text{CP}} \times \underset{(3)}{7890} \times \underset{(4)}{\text{Factor A}}$$

Pumping cost, period II =

$$\text{hp} \times \underset{(1)}{0.746} \times \underset{(2)}{\text{CP}} \times \underset{(3)}{6750} \times \underset{(4)}{\text{Factor A}} \times \underset{(4)}{\text{Factor B}}$$

Pumping cost, period III =

$$\text{hp} \times \underset{(1)}{0.746} \times \underset{(2)}{\text{CP}} \times \underset{(3)}{5080} \times \underset{(4)}{\text{Factor C}} \times \underset{(4)}{\text{Factor D}}$$

(1) Kw/hp; (2) cost of power, \$/Kwhr; (3) hr/yr; (4) present value factors.

Total pumping cost = pumping cost, period I + pumping cost, two-pump operation period II + pumping cost, period III

IIa. Pumping Cost, Present Valued, Combination One- and Two-Pump Operation

One-pump operation at 325 Mw at 38.87 and 45.62 F. 284 Mw at 38.87 and 45.62 F.

Two-pump operation for all other loads and temperatures.

The number of operating hours per year during period I is 700 hr at 325 Mw and 3330 hr at 284 Mw. The circulating water inlet temperatures of 38.87 and 45.62 F are the monthly average temperatures for 4- and 2-mo periods of the year.

The number of operating hours with one pump during period I:

325 Mw:

$$\underset{\text{hr/yr}}{700} \times \underset{\substack{\text{fraction of}\\ \text{year with}\\ \text{38.87 F}\\ \text{water}}}{4/12} + \underset{\text{hr/yr}}{700} \times \underset{\substack{\text{fraction of}\\ \text{year with}\\ \text{45.62 F}\\ \text{water}}}{2/12} = 700\ (0.500)$$

284 Mw:

$$\underset{\text{hr/yr}}{3330} \times \underset{\substack{\text{fraction of}\\ \text{year with}\\ \text{38.87 F}\\ \text{water}}}{4/12} + \underset{\text{hr/yr}}{3330} \times \underset{\substack{\text{fraction of}\\ \text{year with}\\ \text{45.62 F}\\ \text{water}}}{2/12} = 3330\ (0.500)$$

Total number of operating hours with one pump:

$$0.500(700 + 3330) = 4030 \times 0.500 = 2015 \text{ hr.}$$

Based on the characteristic curves of circulating pumps on an existing unit of similar size, the horsepower of one pump operating alone is estimated to be about 0.4 of the horsepower of both pumps operating together. Hence, the equivalent number of hours when using the horsepower of both pumps as a factor will be:

$$2015 \times 0.4 = 806 \text{ hr}$$

The total number of equivalent hours in this case will be:

$$7890 - 2015 + 806 = 6681 \text{ hr}$$

The total number of operating hours per year during period I is 7890.

Similarly for period II:

325 Mw:

$$440\ (\text{hr/yr}) \times 0.500$$

284 Mw:

$$2150\ (\text{hr/yr}) \times 0.500$$

Total number of operating hours with one pump:

$$(2150 + 440)0.500 = 2590 \times 0.500 = 1295 \text{ hr}$$

Equivalent number of hours:

$$0.4 \times 1295 = 520 \text{ hr}$$

The total number of equivalent hours are:

$$6750 - 1295 + 520 = 5975 \text{ hr}$$

The total number of operating hours per year during period II is 6750.

For period III.

325 Mw:

$$0$$

284 Mw:

$$790\ (\text{hr/yr}) \times 0.500$$

Total number of operating hours with one pump:

$$790 \times 0.500 = 395 \text{ hr}$$

Equivalent number of hours:

$$0.4 \times 395 = 158 \text{ hr}$$

Total equivalent hours:

$$5080 - 395 + 158 = 4843 \text{ hr}$$

The total number of operating hours per year during period III is 5080.

Pumping cost, period I =

$$\text{hp} \times \overset{(1)}{0.746} \times \overset{(2)}{\text{CP}} \times \overset{(3)}{6681} \times \overset{(4)}{\text{Factor A}}$$

Pumping cost, period II =

$$\text{hp} \times \overset{(1)}{0.746} \times \overset{(2)}{\text{CP}} \times \overset{(3)}{5975} \times \overset{(4)}{\text{Factor A}} \times \overset{(4)}{\text{Factor B}}$$

Pumping cost, period III =

$$\text{hp} \times \overset{(1)}{0.746} \times \overset{(2)}{\text{CP}} \times \overset{(3)}{4843} \times \overset{(4)}{\text{Factor C}} \times \overset{(4)}{\text{Factor D}}$$

(1) Kw/hp; (2) cost of power, \$/kwhr; (3) hrs/yr; (4) present value factor.

Total pumping cost combination one- and two-pump operation = pumping cost, period I + pumping cost, period II + pumping cost, period III.

### III. Cost of Coal, Present Valued

The cost of coal is to be calculated twice, first for two-pump operation and later for a combination one- and two-pump operation. The cost of coal is to be calculated for each period of each of the four loads, and each of the circulating water inlet temperatures and then totaled as follows:

Cost of coal, present valued = turbine heat rate × kw × hours per year for the respective load and period × fraction of year for the respective circulating water inlet temperature × reciprocal of boiler efficiency × reciprocal of operating efficiency × cost of coal per Btu × present value factor

#### Two-Pump Operation

The condenser vacuum must be known in order to determine the heat rate.

The following formulas and tables are used to calculate the condenser vacuum for two-pump operation:

| Circulating Water Inlet Temp. | Temp. Correction Factor—$F_t$ |
|---|---|
| 38.87 | 0.739 |
| 45.62 | 0.800 |
| 54.50 | 0.881 |
| 65.92 | 0.971 |
| 50.66 | 0.842 |
| 70.00 | 0.993 |

Tube material and gauge correction factor ($F_{tm}$) = 1.000

Tube cleanliness factor ($F_c$) = 0.850 (determined from station records on condenser performance)

VALUES FOR $U_b$-BASIC HEAT TRANSFER RATE

| Tube OD, In. | *1 | 2 | 3 |
|---|---|---|---|
| 0.625 and 0.750 | 687 | 713 | 739 |
| 0.875 and 1.000 | 671 | 696 | 722 |

* Velocity, Fps—[1]6.5 [2]7.0 [3]7.5

$$\text{Corrected heat transfer rate } (U) = U_b \times F_t \times F_{tm} \times F_c$$

$$\begin{array}{l}\text{Logarithmic}\\ \text{mean temp. diff.}\end{array} = MD = \frac{\text{heat rejected per lb} \times \text{condenser flow}}{U \times \text{condenser surface}}$$

Use Table II to determine the heat rejected per pound.

The data on Table II was obtained from Fig. 2. Fig. 2 is the result of cross-plotting the data on Fig. 1.

Use Table III to determine the condenser flow:

$$\text{Temperature rise} = T_r = \frac{\text{Heat rejected per lb} \times \text{condenser flow}}{500 \times \text{circulating water flow, gpm}}$$

$$\text{Range} = \frac{-T_r e^{T_r/MD}}{1 - e^{T_r/MD}}$$

$$\text{Vacuum temp.} = T_s = \text{circulating water inlet temperature} + \text{range}$$

$$\text{Terminal diff} = TD = \text{range} - T_r = T_s - T_2$$

The computer calculated the terminal difference by the range minus the temperature rise since this data was already stored in the machine. The circulating water outlet temperature ($T_2$) was not calculated by the machine.

TABLE I—TYPICAL HEAT RATES AT VARIOUS BACK PRESSURES AND LOADS

(Btu per kwhr)

| Pressure, In. Hg | 325 Mw | 284 Mw | 203 Mw | 144 Mw |
|---|---|---|---|---|
| 0.3 | 7536 | 7506 | 7469 | 7515 |
| 0.4 | 7527 | 7507 | 7498 | 7568 |
| 0.5 | 7521 | 7512 | 7524 | 7623 |
| 0.6 | 7521 | 7520 | 7559 | 7673 |
| 0.7 | 7523 | 7530 | 7590 | 7723 |
| 0.8 | 7530 | 7542 | 7621 | 7770 |
| 0.9 | 7539 | 7556 | 7652 | 7820 |
| 1.0 | 7550 | 7573 | 7682 | 7865 |
| 1.1 | 7562 | 7591 | 7716 | 7911 |
| 1.2 | 7577 | 7612 | 7748 | 7954 |
| 1.3 | 7594 | 7634 | 7783 | 7997 |
| 1.4 | 7613 | 7658 | 7814 | 8040 |
| 1.5 | 7635 | 7683 | 7846 | 8086 |
| 1.6 | 7657 | 7706 | 7877 | 8132 |
| 1.7 | 7679 | 7728 | 7908 | 8177 |
| 1.8 | 7701 | 7750 | 7938 | 8220 |
| 1.9 | 7722 | 7770 | 7968 | 8262 |
| 2.0 | 7742 | 7789 | 7997 | 8300 |
| 2.1 | 7762 | 7809 | 8026 | 8339 |
| 2.2 | 7781 | 7827 | 8054 | 8374 |
| 2.3 | 7800 | 7846 | 8082 | 8408 |
| 2.4 | 7819 | 7865 | 8108 | 8441 |
| 2.5 | 7837 | 7888 | 8135 | 8474 |

These calculations are from correction factors given in the turbine contract for this turbine.

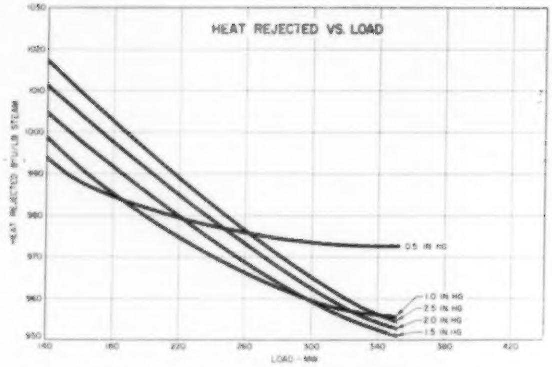


**Fig. 1—The above family of curves depict the heat leaving in the steam over a range of load when the turbine discharges against certain back pressures**

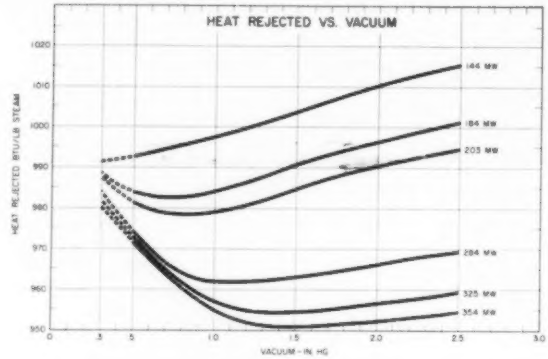


**Fig. 2—The information in Fig. 1 has been replotted to picture the heat leaving in the steam over a range of back pressures but under specific turbine loads**

If $TD$ is less than 5 F then correct $T_s$ as follows:

$$T_s = \text{circulating water inlet temperature} + 5 + T_r$$

The corresponding vacuum at the temperature $T_s$ is listed on Table IV.

If the calculated vacuum is within 0.01 in. Hg of the assumed vacuum the turbine heat rate can be obtained from Table I.

The turbine heat rate data on Table I was determined by applying the vacuum correction factors to the contract or given data at 1 in. Hg.

CONTRACT OR GIVEN DATA AT 1 IN. Hg

| Mw | 325 | 284 | 203 | 144 |
|---|---|---|---|---|
| Throttle flow, lb per hr | 2,070,000 | 1,770,000 | 1,225,000 | 830,000 |
| Gross turbine heat rate, Btu per kwhr | 7,550 | 7,573 | 7,682 | 7,865 |

TABLE II—TYPICAL HEAT REJECTED TO CONDENSER
(Btu per lb of steam)

| Pressure, In. Hg | 325 Mw | 284 Mw | 203 Mw | 144 Mw |
|---|---|---|---|---|
| 0.3 | 982.0 | 984.3 | 988.2 | 991.5 |
| 0.4 | 977.0 | 979.0 | 984.3 | 992.0 |
| 0.5 | 972.4 | 974.0 | 981.5 | 992.6 |
| 0.6 | 968.5 | 969.8 | 979.6 | 993.4 |
| 0.7 | 964.7 | 966.4 | 978.6 | 994.4 |
| 0.8 | 961.5 | 963.2 | 978.3 | 995.4 |
| 0.9 | 958.7 | 962.3 | 978.4 | 996.4 |
| 1.0 | 956.6 | 961.7 | 979.0 | 997.5 |
| 1.1 | 955.4 | 961.6 | 979.8 | 998.5 |
| 1.2 | 954.6 | 961.8 | 980.8 | 999.6 |
| 1.3 | 954.3 | 962.1 | 982.0 | 1000.8 |
| 1.4 | 954.2 | 962.4 | 983.3 | 1002.1 |
| 1.5 | 954.4 | 962.8 | 984.7 | 1003.3 |
| 1.6 | 954.6 | 963.5 | 986.0 | 1004.7 |
| 1.7 | 955.0 | 964.0 | 987.2 | 1006.0 |
| 1.8 | 955.6 | 964.6 | 988.3 | 1007.3 |
| 1.9 | 956.0 | 965.3 | 989.4 | 1008.7 |
| 2.0 | 956.7 | 966.0 | 990.3 | 1010.0 |
| 2.1 | 957.1 | 966.7 | 991.2 | 1011.2 |
| 2.2 | 957.4 | 967.4 | 992.1 | 1012.3 |
| 2.3 | 958.0 | 968.0 | 992.9 | 1013.3 |
| 2.4 | 958.6 | 968.7 | 993.7 | 1014.3 |
| 2.5 | 959.3 | 969.3 | 994.6 | 1015.3 |

These rejection rates were worked up from figures supplied by the turbine engineers. Their figures were plotted against the back pressures supplied and then cross-plotted to obtain the rejection rates at the loads set in the scope of the work. (See Figs. 1 and 2.)

VACUUM CORRECTION FACTORS
PER CENT OF HEAT RATE

| | Throttle Flow, Lb per Hr | | | |
|---|---|---|---|---|
| Exhaust pressure, in. Hg | 2,330,000 | 2,000,000 | 1,500,000 | 1,000,000 |
| 0.5 | −0.20 | −0.45 | −1.40 | −2.60 |
| 1.0 | 0 | 0 | 0 | 0 |
| 1.5 | 1.00 | 1.20 | 1.75 | 2.50 |
| 2.0 | 2.40 | 2.60 | 3.35 | 4.85 |
| 2.5 | 3.75 | 3.85 | 4.80 | 6.90 |

The data in the vacuum correction factor table is plotted to obtain Fig. 3. The turbine engineers throttle flow at 325, 284, 203, and 144 Mw at 1 in. Hg is drawn

TABLE III—TYPICAL CONDENSER FLOW—TOTAL STEAM
(Lb per hr)

| Pressure, In Hg | 325 Mw | 284 Mw | 203 Mw | 144 Mw |
|---|---|---|---|---|
| 0.3 | 1,369,417 | 1,194,598 | 864,384 | 621,580 |
| 0.4 | 1,376,388 | 1,200,679 | 868,784 | 624,744 |
| 0.5 | 1,382,945 | 1,206,399 | 872,923 | 627,720 |
| 1.0 | 1,410,000 | 1,230,000 | 890,000 | 640,000 |
| 1.5 | 1,429,768 | 1,247,245 | 902,478 | 648,973 |
| 2.0 | 1,444,334 | 1,259,951 | 911,672 | 655,584 |
| 2.5 | 1,451,623 | 1,266,310 | 916,273 | 658,893 |
| 3.0 | 1,463,072 | 1,276,297 | 923,500 | 664,090 |
| 3.5 | 1,468,275 | 1,280,836 | 926,784 | 666,451 |

TABLE IV—PRESSURE CORRESPONDING TO $T_s$

| $T_s$ F | Pressure, In. Hg | $T_s$ F | Pressure, In. Hg |
|---|---|---|---|
| 42 | 0.2677 | 86 | 1.2527 |
| 44 | 0.2892 | 88 | 1.3347 |
| 46 | 0.3122 | 90 | 1.4215 |
| 48 | 0.3365 | 92 | 1.5131 |
| 50 | 0.3627 | 94 | 1.6097 |
| 52 | 0.3907 | 96 | 1.7117 |
| 54 | 0.4203 | 98 | 1.8192 |
| 56 | 0.4522 | 100 | 1.9325 |
| 58 | 0.4858 | 102 | 2.0519 |
| 60 | 0.5218 | 104 | 2.1775 |
| 62 | 0.5601 | 106 | 2.3099 |
| 64 | 0.6009 | 108 | 2.4491 |
| 66 | 0.6442 | 110 | 2.5955 |
| 68 | 0.6903 | 112 | 2.7494 |
| 70 | 0.7392 | 114 | 2.9111 |
| 72 | 0.7912 | 116 | 3.0806 |
| 74 | 0.8462 | 118 | 3.2589 |
| 76 | 0.9046 | 120 | 3.4458 |
| 78 | 0.9666 | 122 | 3.6420 |
| 80 | 1.0321 | 124 | 3.8475 |
| 82 | 1.1016 | 126 | 4.0600 |
| 84 | 1.1750 | | |

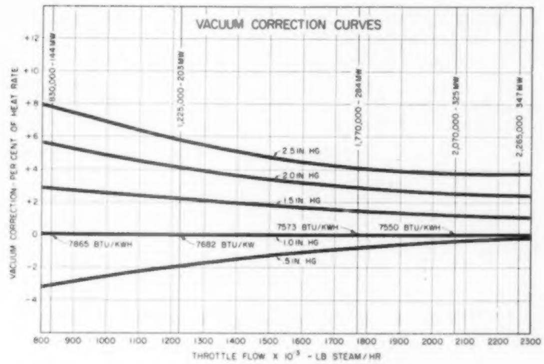


**Fig. 3—The data in the vacuum correction factor table appearing in the text has been plotted. With the turbine throttle flows at specific loads (347-, 325-, 284-, 203- and 144-Mw) the vacuum correction as a per cent of heat rate against the 1-in. Hg base can be picked off at these values for different back pressures**

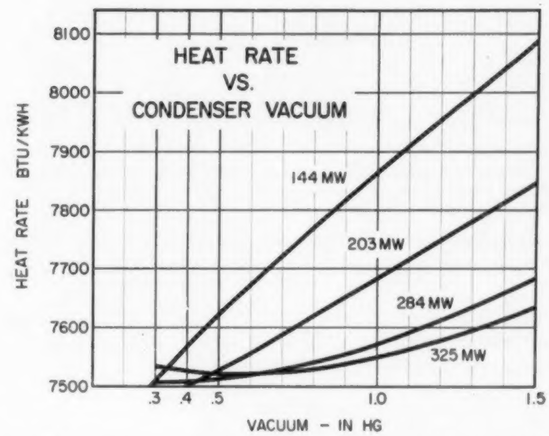


**Fig. 4—Heat rate curve was made up from the data in Fig. 3 so that it depicts a constant load at the various back pressures. Actually the heat rate data was obtained for constant throttle flow at the back pressures. The heat rate is correct at 1 in. Hg but requires slight modifications as the back pressure deviates**

as a vertical line on Fig. 3. The vacuum correction as per cent of heat rate can now be read from Fig. 3 at 0.5, 1.0, 1.5, 2.0 and 2.5-in. Hg exhaust pressures at the 1-in. Hg throttle flows. Using these corrections the turbine heat rate at this throttle flow can be calculated for 0.5, 1.0, 1.5, 2.0, and 2.5 in. Hg. This data is plotted on Fig. 4 as heat rate versus vacuum with throttle flow at the load points as a parameter.

While this article was being written it was realized that the heat rate curve, Fig. 4, was drawn for constant load at the various back pressures, whereas the heat rate was actually obtained for constant throttle flow at the various back pressures. The heat rate is correct at 1 in. Hg but needs some additional modification as the vacuum deviates from that pressure. Since the hours of operation of the unit was set up on the basis of constant load versus back pressure, the turbine specifications should be written to obtain the vacuum correction factors on the basis of constant load. It is believed that for this particular study, the circulating water inlet temperature was low enough that the condenser selection was uneffected.

The data on Table V, VI and VII is used to determine the remaining items in the cost of coal calculation:

Example:
Cost of coal for period I at 325 Mw, 38.87 F circulating water inlet temperature
Cost of coal =
Turbine heat rate (from Table I at the calculated vacuum) × 325,000 (kw) × 700 (hr/yr, Table V)
× 0.3333 (fraction of year, Table V) × 1/0.8941 (1/blr. eff. Table V) 1/0.93 (operating eff., Table V)
× CC-1 (cost of coal per Btu) × Factor A (present value factor)

*Combination One- and Two-Pump Operation*

For the combination of one- and two-pump operation the vacuum should be calculated in the same manner except that for the two loads and two circulating water inlet temperatures of the one-pump operation. For one-pump operation the circulating water flow will be approximately 65 per cent of flow with two-pump operation. The water velocity through the tubes will be decreased proportionately for one-pump operation.

The basic heat transfer rate table and temperature rise formula must be modified. All the other formulas and tables will remain the same.

TABLE V—COAL COST, PERIOD I, 10 YR

| River Temp., F | Load, Mw | Hr/Yr | Fraction of Year | 1/Blr. Eff. | 1/Op. Ratio |
|---|---|---|---|---|---|
| 38.87, Jan., Feb., March, Dec. | 325 | 700 | 0.3333 | 1/0.8941 | 1/0.93 |
| | 284 | 3330 | " | 1/0.8953 | " |
| | 203 | 2890 | " | 1/0.8975 | " |
| | 144 | 970 | " | 1/0.8990 | " |
| 45.62, Apr., Nov. | 325 | 700 | 0.1666 | 1/0.8941 | 1/0.93 |
| | 284 | 3330 | " | 1/0.8953 | " |
| | 203 | 2890 | " | 1/0.8975 | " |
| | 144 | 970 | " | 1/0.8990 | " |
| 54.50, May June, Oct. | 325 | 700 | 0.2500 | 1/0.8941 | 1/0.93 |
| | 284 | 3330 | 0.2500 | 1/0.8953 | " |
| | 203 | 2890 | " | 1/0.8975 | " |
| | 144 | 970 | " | 1/0.8990 | " |
| 65.92, July, Aug., Sept. | 325 | 700 | 0.2500 | 1/0.8941 | 1/0.93 |
| | 284 | 3330 | " | 1/0.8953 | " |
| | 203 | 2890 | " | 1/0.8975 | " |
| | 144 | 970 | " | 1/0.8990 | " |

Coal cost/Btu = CC-1. Present value = Factor A.

TABLE VI—COAL COST, PERIOD II, 10 YR

| River Temp., F | Load, Mw | Hr/Yr | Fraction of Year | 1/Blr. Eff. | 1/Op. Ratio |
|---|---|---|---|---|---|
| 38.87, Jan., Feb., March, Dec. | 325 | 440 | 0.3333 | 1/0.8941 | 1/0.93 |
| | 284 | 2150 | " | 1/0.8953 | " |
| | 203 | 2670 | " | 1/0.8975 | " |
| | 144 | 1490 | " | 1/0.8990 | " |
| 45.62, Apr., Nov. | 325 | 440 | 0.1666 | 1/0.8941 | 1/0.93 |
| | 284 | 2150 | " | 1/0.8953 | " |
| | 203 | 2670 | " | 1/0.8975 | " |
| | 144 | 1490 | " | 1/0.8990 | " |
| 54.50, May, June, Oct. | 325 | 440 | 0.2500 | 1/0.8941 | 1/0.93 |
| | 284 | 2150 | " | 1/0.8953 | " |
| | 203 | 2670 | " | 1/0.8975 | " |
| | 144 | 1490 | " | 1/0.8990 | " |
| 65.92, July, Aug., Sept. | 325 | 440 | 0.2500 | 1/0.8941 | 1/0.93 |
| | 284 | 2150 | " | 1/0.8953 | " |
| | 203 | 2670 | " | 1/0.8975 | " |
| | 144 | 1490 | " | 1/0.8990 | " |

Coal cost/Btu = CC-2. Present value = Factor A × Factor B.

VALUE FOR $U_B$—BASIC HEAT TRANSFER RATE

(Btu per sq ft per F per hr)

| Tube OD, In. | †1 | 2 | 3 |
|---|---|---|---|
| 0.625 and 0.750 | 556 | 576 | 580 |
| 0.875 and 1.00 | 540 | 561 | 596 |

† Velocity, Fps, One-pump Operation [1]4.225*, [2]4.550, [3]4.875

* The velocities of 4.225, 4.550 and 4.875 were calculated on the basis of 65 per cent of the two-pump operation velocities of 6.5, 7.0, and 7.5 fps.

$$\text{Temperature rise (For one-pump operation)} = \frac{\text{heat rejected per lb} \times \text{condenser flow}}{0.65 \times 500 \times \text{circulating water flow, gpm (two-pump operation)}}$$

The coal cost calculation for combination one- and two-pump operation is the same as for two-pump operation when the above modifications are added.

## IV. Capability Evaluation

Capability evaluation = capability credit for turbine + capability charge for circulating water pump power requirements

### *A. Capability Credit for Turbine*

Table VIII shows the turbine capability used in this study in 0.1-in. Hg increments. These figures were calculated from the vacuum correction data and the following formula:

$$\text{kw} = \frac{\text{maximum capability at 1 in. Hg}}{100 - \text{per cent change in heat rate}}$$

TABLE VII—COAL COST, PERIOD III, 15 YR

| River Temp., F | Load Mw | Hr/Yr | Fraction of Year | 1/Blr. Eff. | 1/Op. Ratio |
|---|---|---|---|---|---|
| 38.87, Jan., Feb., March, Dec. | 325 | 0 | 0.3333 | 1/0.8941 | 1/0.93 |
| | 284 | 790 | " | 1/0.8953 | " |
| | 203 | 1840 | " | 1/0.8975 | " |
| | 144 | 2450 | " | 1/0.8990 | " |
| 45.62, Apr., Nov. | 325 | 0 | 0.1666 | 1/0.8941 | 1/0.93 |
| | 284 | 790 | " | 1/0.8953 | " |
| | 203 | 1840 | " | 1./08975 | " |
| | 144 | 2450 | " | 1/0.8990 | " |
| 54.50, May, June, Oct. | 325 | 0 | 0.2500 | 1/0.8941 | 1/0.93 |
| | 284 | 790 | " | 1/0.8953 | " |
| | 203 | 1840 | " | 1/0.8975 | " |
| | 144 | 2450 | " | 1/0.8990 | " |
| 65.92, July, Aug., Sept. | 325 | 0 | 0.2500 | 1/0.8941 | 1/0.93 |
| | 284 | 790 | " | 1/0.8953 | " |
| | 203 | 1840 | " | 1/0.8975 | " |
| | 144 | 2450 | " | 1/0.8990 | " |

Coal/Btu = CC-3. Present value = Factor C × Factor D.

In this study:

$$\text{kw at 0.5 in. Hg.} = \frac{347{,}000}{100 - 0.32} = 348{,}110 \text{ kw}$$

Capability credit at 100 per cent and 0.5 in. Hg = (computed kw − Base kw at 2.5 in. Hg) × PC, $ per kw plant × Factor "X" present value factor

= (348,100 − 334,393 × PC × Factor "X"

Capability credit at 90 per cent and 0.5 in. Hg = 0.90 (348,100 − 334,393) × PC × F "X"

### *B. Capability of Circulating Water Pump Power Requirements*

Capability charge for circulating water pump power at 90 and 100 per cent = pump hp × PC × Factor "X"

## V. Total Cost, Present Valued, Two-Pump Operation Only

The total cost is calculated by the following:

$$\text{V} = \text{I} + \text{II} + \text{III} - \text{IV}$$

## Va. Total Cost, Present Valued, Combination One- and Two-Pump Operation

The total cost is calculated by the following:

$$\text{Va} = \text{I} + \text{IIa} + \text{IIIa} - \text{IV}$$

TABLE VIII—TYPICAL CAPABILITY AT VARIOUS BACK PRESSURES

| Pressure, In. Hg | Kw | Pressure, In. Hg | Kw |
|---|---|---|---|
| 0.5 | 348,110 | 1.5 | 343,190 |
| 0.6 | 348,000 | 1.6 | 342,300 |
| 0.7 | 347,900 | 1.7 | 341,390 |
| 0.8 | 347,700 | 1.8 | 340,400 |
| 0.9 | 347,450 | 1.9 | 339,450 |
| 1.0 | 347,000 | 2.0 | 338,536 |
| 1.1 | 346,300 | 2.1 | 337,600 |
| 1.2 | 345,600 | 2.2 | 336,700 |
| 1.3 | 344,850 | 2.3 | 335,900 |
| 1.4 | 344,000 | 2.4 | 335,100 |
| Base 2.5 | 334,393 | | |

## General Electric Announces New Power Industry Service

J. F. Young, manager of G-E's Electric Utility Systems Engineering and Planning Operation announced the appointment of Charles C. Thomas to the post of manager of a newly formed Power Plant Engineering Operation.

This new operation was set up to concentrate and integrate General Electric skills and efforts in the engineering of modern power plants, in order to better serve the electric utility industry and the industry's consulting engineers.

It was conceived to meet a growing international business participation, repeated foreign cometptition and the growing pace of technology.

It will not try to take the place of engineering firms, architects, contractors or utility organizations which do their plant engineering themselves.

But it will try to bring General Electric's technology together in working with consulting engineers and utilities to better optimize power plant designs where new manufacturing technology can make a contribution. It will serve in a partnership role and aid the International General Electric in developing turnkey propositions and in handling turnkey contracts.

It will aid in nuclear power plants where it can, but responsibility for nuclear power plants is in the General Electric's Atomic Power Equipment Department.

Regarding conventional plants, aid will be provided in special situations through technology or new opportunities for innovations where it is indicated that General Electric can make a contribution.

By IGOR J. KARASSIK*

Worthington Corp.

**The boiler feed pump and its associated equipment represent a major operating and maintenance consideration in today's power plant. Here we run in question and answer form a series of clinic sessions on various boiler feed pump problems. The replies are the work of one of the topmost pump authorities and give specific information which we hope will prove valuable to our readers.**

# Steam Power Plant Clinic—Part XVII

QUESTION

*I have heard conflicting statements regarding the effect of temperature on the required NPSH for boiler feed pumps. Some engineers claim that more NPSH is required at higher temperatures and produce as evidence a chart published in the Standards of the Hydraulic Institute which shows margins to be added to the recommended NPSH—margins which increase with increasing temperatures. On the other hand, I have heard recently that less NPSH is required when handling hot water than with cold water. Which of these statements is correct?*

ANSWER

Oddly enough, both statements are correct if they are properly interpreted. If we refer strictly to steady state conditions, the required NPSH is reduced with increasing temperatures. The theory underlying this effect is fairly simple but need not be discussed in detail here. It is based on the fact that mild and partial cavitation can take place in a pump without causing extremely unfavorable effects. The higher the temperature, the less volume will be occupied by the steam into which a small amount of water will flash if NPSH is reduced until cavitation just begins. Thus, the same amount of pressure reduction below the vapor pressure will liberate less steam volume and will reduce the pump effective capacity by a lesser amount. A very complete treatment of this theory and of supporting test evidence is available in a recent ASME paper.[1]

This effect can also be demonstrated by strictly logical considerations. Suppose that we are considering the effect of temperature on the performance of centrifugal pumps, operating at a given speed and supplied with a fixed NPSH, when handling: (a) cold water at 70 F and (b) hot water at 705.4 F (critical temperature for water).

* Consulting Engineer and Manager of Planning, Harrison Div.

It is obvious that in the case of cold water at 70 F, the break in the head-capacity curve will occur at some capacity determined by the geometry of the impeller and by the speed at which the pump is operating as indicated on Fig. 1.

When it comes to handling water at 705.4 F, that is the critical temperature at which no evaporation takes place (as long as the pressure is maintained at 3206.2 psia), the volume occupied by water is the same as that occupied by steam. Under these conditions, a slight reduction in the available NPSH can have no appreciable effect on the performance characteristics of the pump (when expressed in foot-pounds per pound, plotted against volume) since no true cavitation can take place. This statement is predicated on the fact that a change from water to steam takes place without a change in volume occupied by either fluid. In other words, the head-capacity curve of a pump handling 705.4 F water

[1] "Cavitation and NPSH Requirements of Various Liquids" by Victor Salemann, ASME Paper 58-A-52, presented at the ASME Annual Meeting, December 1–5, 1958, New York City.

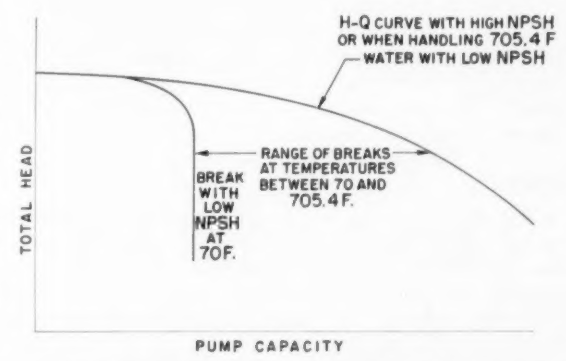


**Fig. 1—Effect of temperature on maximum capacity at fixed pump speed and with fixed NPSH**

with less than the theoretically required NPSH will coincide with the head-capacity curve with cold water and ample NPSH, as shown on Fig. 1.

Having established the two limits of performance characteristics at 70 and at 705.4 F, we can assume that the relationship between the location of the break and the pumping temperature is a continuous function. Therefore, all breaks at temperatures between these two extreme temperatures must take place between the two limits of capacity indicated.

This would indicate that the practice of assuming that the required NPSH is independent of the pumping temperature is conservative and unrealistic, and that this NPSH is definitely reduced with increasing temperatures.

On the other hand, the margin recommended to be added by the Standards of the Hydraulic Institute takes into consideration actual conditions which prevail in a steam power plant—which differ considerably from the "steady state" conditions we assumed in establishing our relation between NPSH requirements and operating temperature. It is well known that "transient conditions" such as sudden load reductions will introduce unfavorable effects on the suction conditions in an open cycle, where the boiler feed pump takes its suction from a deaerating direct contact heater. A very severe reduction in available NPSH follows the sudden load reduction (see COMBUSTION, Aug. 1959 and May 1960, "Steam Power Plant Clinic").

Therefore, means must be employed to take care of the time lag which exists between the instantaneous reduction of pressure in the heater which follows a sudden load drop and the ultimate reduction of temperature at the pump suction, after the feedwater already in the suction piping will have been pumped out into the discharge header. The most logical solution is to provide a factor of safety for the installation through the addition of some arbitrary amount to the required NPSH under steady state conditions.

The degree of NPSH reduction caused by load reduction is intimately connected with the operating pressure in the heater—and, hence, with the operating temperature: the higher this pressure the more severe the NPSH reduction.

The chart in the Standards of the Hydraulic Institute is therefore intended to compensate, at least partially, for this effect. The margin indicated is purely empirical and does not necessarily cover all cases, since in addition to the operating temperature several other factors enter into establishing the amount by which the available NPSH will be reduced after a sudden load reduction. Some of these factors are the storage volume in the heater, the configuration and internal volume of the suction piping and the behavior of the feedwater regulator following the sudden load reduction. Thus, in some cases, the recommended margin may be insufficient while in others it can be excessive. The merit of the chart lies in bringing attention to the importance of giving this problem consideration.

Returning now to the thought that under steady state conditions required NPSH values are reduced with increased temperatures, it is necessary to introduce a word of caution: it would be highly unwise to rush headlong to the conclusion that the values of recommended NPSH for boiler feed pumps should be drastically reduced. We must remember that boiler feed pumps are frequently required to operate over wide ranges of temperature and that a given installation cannot very well take advantage of a permissible reduction in NPSH requirements at the top operating temperature if operation at lower temperatures will wipe out any assistance gained by the effect we have described. The effect is also very useful in affording us additional protection during the transient conditions which prevail when severe load reduction occurs in a steam power plant. Let us preserve this protection and not give in to the temptation to cut corners.

QUESTION

*I have noticed that the rated capacity of condensate pumps is generally selected with considerably more margin over the maximum expected condensing steam flow than is the case with boiler feed pumps when their rated capacity is compared to the maximum turbine throttle flow. When I say more margin, I really mean a considerable difference. For instance, I have seen cases where as much as 30 per cent or more is added to the maximum condensing steam flow in selecting condensate pumps. Is this difference influenced by the thought that condensate pumps may operate under cavitating conditions and therefore their effective capacity will be reduced much below their rated capacity?*

ANSWER

The effect of cavitation is not the determining factor in selecting condensate pump capacity. To begin with, the use of cavitation to control the delivery of a condensate pump (this is generally referred to as "submergence control" or "self-regulation") is restricted to open feedwater systems, where the condensate pump delivers into a direct contact heater. While this arrangement used to be quite popular some time ago, it is less frequently used today and the delivery of the condensate pumps is controlled by a throttling valve

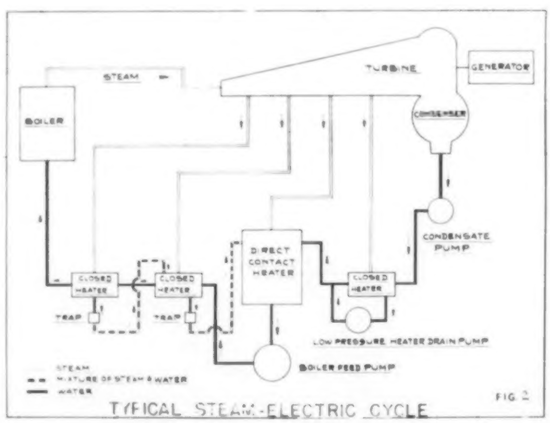


**Fig. 2—Typical present day steam-electric cycle employs a heater drain pump to handle the difference between condensed steam flow and the flow required from the bf pump**

which takes its impulse from either the condenser hotwell level or the level in the storage space of the direct contact heater.

Moreover, even when condensate pumps are operated on submergence control, their submergence requirements are so chosen that they can deliver their full rated capacity under prevailing submergence conditions and no limitation on their effective capacity is allowed to be imposed by the static elevation conditions of the installation.

Condensate pumps must be designed for a considerably higher capacity than they normally will have to handle because of the very nature of the usual feedwater cycle. If you refer to Fig. 2, you will see that the typical regenerative cycle includes a series of closed heaters located between the condensate pump and the boiler feed pump. Generally, the heater drains from these heaters are cascaded down to the lowest pressure heater in the system, from which they are pumped back into the condensate discharge header by a heater drain pump. In this manner, the heater drain pump handles the difference between the condensed steam flow and the flow required from the boiler feed pump.

In the event abnormal conditions arise, as for instance the necessity for bypassing the closed heaters or the unavailability of the heater drain pump, a condensate pump designed for just the normal condensing steam flow would be inadequate. If the heaters are bypassed and no extraction steam is taken from the bleed stages, the condensing steam flow will be much increased over its normal value and the condensate pump must be capable of evacuating the hotwell.

If, on the other hand, the heater drain pump is unavailable for some reason, the heater drains will be diverted and dumped into the condenser hotwell, swelling the flow from the turbine exhaust. In this case, as well, it becomes necessary to have sufficient capacity built into the condensate pump to handle all the flow that is entering the condenser hotwell.

Thus, whereas the boiler feed pump is designed for something like 8 to 15 per cent excess capacity—and that only to take care of boiler swings and of pump wear—condensate pumps may easily be rated for up to 30 per cent in excess of normal condensate steam flow.

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## Electric Power Experts Joins Computer Field

J. H. (Jeff) Hunnicutt has been appointed to the position of applications engineer specializing in computer control and monitoring systems for the electric power industry, Henry L. Bechard, products manager of the Thompson-Ramo-Wooldridge Products Co., announced recently.

This new position results from the increased emphasis on power industry applications of digital control computers and is an outgrowth of the cooperative agreement between TRW Products and Republic Flow Meters Co. to design and furnish complete computer-operated systems for electric power generating stations.

Prior to joining TRW Products in February, Hunnicutt was an associate editor of *Power Engineering* magazine. He was formerly a startup and plant betterment engineer with Stone & Webster Engineering Corp. of Boston and United Engineers and Constructors, Inc., of Philadelphia.

# American Power Conference in Review—II

LAST month's issue of COMBUSTION carried a fair portion of the American Power Conference papers held to be of value to the readers of this magazine. There were, however, so many that in the interest of space the complete meeting report was divided into two parts and the majority was held over for this issue.

#### Condensers

One of the more comprehensive treatments of condensers and their auxiliaries was featured at the Conference. **Rushen A. Wilson,** Allis-Chalmers Mfg. Co., handled the subject of "Some Hydraulic Aspects of Condenser Circulating Water Systems." The circulating water systems, the authors declared, as well as all other equipment in modern power plants have been getting larger and more complicated. These water systems differ from other hydraulic systems in many respects, one of which is the presence of vacuum piping. Efficiency and low operating power are important in circulating water systems, but dependability and reliability are of equal importance. The air lift pump is an old type consisting of air pipe and an eductor tube going down a proper distance below the water in a well, with the eductor tube discharging above ground This principle of the air or vapor lift can cause trouble in vacuum lines. It may make an air ejector intercooler drain loop seal flash over if there is an air leak on the condenser side of the loop. It may prevent drainage if there is one on the intercooler side.

A vacuum will not lift pure water more than 34 feet, but this does not apply to a mixture of air and water. This should be considered when running vent lines from pipes or water boxes to vacuum pumps. Preferably there should be a vent separating tank, with an adequate drain, between vessels being vented and any mechanical vacuum pumps. Hydraulic air compressors are essentially the reverse of air lift pumps. The authors showed a good engineering siphon that was reliable had proved itself in all sizes and could stand a considerable variation in flow. R. T. Richards' papers, the author felt, before the American Society of Civil Engineers in 1957 were the first to mention that these arrangements did have quite different venting characteristics. These differences have been well known by individual engineers.

There have been cases where such siphons caused trouble in parallel installations, in which two or more condensers were supplied by a common supply pipe or tunnel, or where the two sides of a divided water box condenser were so supplied. The solution to this is to install a common vent between all parallel siphons. All parallel siphons should be vented together at the high point, even though it is true that hundreds have operated for years without common vents.

Further the author believes you should keep the discharge from air-operated primers or vacuum pumps out of discharge tunnels and it becomes necessary to keep it out if there is a slanting invert anywhere downstream. Air present in appreciable quantities can give an air lift pump effect that can be mistaken for some mysterious water hammer.

Siphon piping well designed with no slanting piping is self venting. With good design and an apparently good separate venting system about 1 cfm per 20,000 gpm of air can be liberated and removed by the venting system. Venting systems should be able to remove the air from the water space of a divided water box condenser after one half has been shut down. The same applies for the individual condensers of an installation served by one pressurized tunnel. There have been a few units where a vent system or vacuum pump was required to start the unit, but most pumps will push sufficient water over the high point to get started.

The presence of air should not cause or promote water hammer but might even be a cushion to minimize it. The presence of vapor in the system at start-up is a different matter and is considered dangerous. Water jets, steam jets, air jets and all kinds of mechanical vacuum pumps have been used to prime and vent circulating water systems.

**W. E. Glausser, C. H. Unruh** and **G. Balekjian,** C. F. Braun & Co., described the "Plant Test of a 95,000-Square Foot-Condenser." Unit One of the Scattergood Station of the L. A. Water and Power Dept. went on the line in December 1958. Extensive tests were made during the first three months of operation to evaluate the performance of the condenser and the authors believed that the scope of the test data reported is greater than any previously reported literature.

The 95,000 sq ft condenser under test is a single pass with $^7/_8$-in., 18-gage, 40-ft long tubes of 70–30 copper-nickel, rated for 1.5 in. of mercury back-pressure using 78,000 gpm of sea water at 60 F. Design water-velocity is 5.1 ft per sec, and the design cleanliness-factor is 75 per cent. The tubesheets are 70–30 copper-nickel.

The first part of the authors paper dealt with condenser performance, the second part with test equipment and the testing procedures, and the last part with an over-

all evaluation of the test data and condenser performance. We have confined our abstract remarks to the first part of the paper.

Steam back-pressure at the condenser inlet, as the authors indicated, must be measured accurately to compute the overall HEI heat-transfer coefficient. Ten pressure-taps were installed by General Electric in the turbine outlet-duct. They are located about one foot above the expansion joint, and at various points in the duct cross-section. The control room pressure-tap is located just below the expansion joint. Braun installed four taps at the top of the tubefield near the center steam-lane. Three taps are on the south side—one in the middle and one near each end. The fourth tap is at the middle of the north side. At each of the four pressure-taps at the top of the tubefield, Braun installed thermocouples at the same locations. As might be expected, back pressures measured at the different points were not entirely consistent. Saturation pressures corresponding to the measured temperatures are more reliable than the pressure measurements for two reasons. First, the measurement of temperature is inherently more accurate at these conditions than the measurement of pressure. Pressure measurement is reliable to about plus or minus 0.03 in. of mercury which is equivalent to about plus or minus 0.6 deg F at design conditions. Temperature measurements are reliable to about plus or minus 0.3 deg F. Second, temperature measurement is not affected by velocity-head effects caused by localized turbulence of the high-velocity stream.

Prediction of steam pressure-drop through the tube-banks is an important part of surface-condenser design. Pressure drop reduces the condensing temperature, thereby decreasing the thermal driving force for heat transfer. On the other hand, a certain amount of pressure drop is necessary to get good flow distribution and eliminate possible stagnant areas where noncondensables can collect.

A summary of the tubebank pressure-drop profiles at 100 per cent steam-load was given in table form. Some interesting observations can be made by comparing the cold end with the hot end pressure-drop. Most of the pressure drop at the cold end occurs in the main tube-field. Most of the pressure drop at the hot end occurs in the air-cooling section.

Heat transfer was measured and computed in two ways. The overall coefficient was determined for the entire condenser based on the steam inlet-temperature and the overall cooling-water temperatures. The second set of heat-transfer data were taken for individual tubes in the condenser. The individual-tube data were taken at various water velocities and steam loads. The condenser is equipped with 40 tubes which can be individually tested. They are arranged in groups of four tubes each, in five areas of each half of the tubefield.

The authors remarked that cooling water was circulated in all but the test tubes of the condenser for about two weeks prior to the time the plant was started up. For this reason, the main bulk of the condenser tubes had gone through the initial fouling period before any testing was started. All the overall heat-transfer data reflect this fouled condition.

From all observations, it seemed to the authors, to be much better to install a thermocouple in the neck of the condenser, rather than a pressure-tap, to determine condenser back-pressure. The steam to the condenser is wet. Therefore, they reasoned, the temperature will always be the saturation temperature corresponding to the absolute pressure. The thermocouple reading will not be influenced by the velocity of the steam in the throat.

"Economic Sizing of Condensers Through the Use of the Digital Computer" by **M. J. Archbold, Frank V. Miholits, Ami Leidner** and **C. E. Person,** Commonwealth Edison Co., will be run in full in COMBUSTION, May and June issues. As the authors very early stated, to properly size a condenser consideration must be given to using many variables such as tube lengths and diameters, heat transfer surfaces, and water velocities, as well as turbine loads and circulating water temperatures. To calculate just a few condenser sizes requires a very large amount of labor and a certain amount of short cutting to bring the work within practical bounds. Even though the work is conscientiously done, the size selection is often under question.

The authors felt that for the large units being installed, the longhand evaluation is no longer satisfactory. For example, for a large (300 Mw) unit even a small improvement in condenser vacuum would result in a considerable savings in operating cost during the 35 year life span of the unit, and hence a more rigorous and refined evaluation should be employed in determining the optimal condenser size. It was for this reason that the Commonwealth Edison Co. made use of a digital computer in the selection of several of their recent condensers.

*Editor's Note: Because the use of this computer in a specific power plant problem combines so much interesting design considerations checked out by a relatively new design aid, the digital computer, we elected to run this paper in full in the May, June issues of the publication.*

"Performance Tests of Water Lubricated Bearing Materials for Condenser Circulating Water Pumps" was presented by **F. L. Yetter** and **C. H. Diehl,** C. H. Wheeler Mfg. Co. For many years the design factors and materials used for the bearings and journals in the various kinds of pumps followed the general rules for standard materials used in the machinery industry. Pump service conditions have become so much more severe with heavier bearing loads and higher service speeds, a number of investigations have been undertaken by groups such as Battelle Memorial Institute, Franklin Institute and several manufacturers of bearings to determine the design limits and newer materials that could be applied to these more severe conditions.

The test apparatus in the program carried on at C. H. Wheeler Mfg. Co. was designed to provide a practical test comparable to the actual service of bearing-journal in a pump; thus the test results will reflect the performance of the prototype under field conditions. The apparatus is a departure from the usual methods of "wear testing," but it does provide the operating conditions experienced in the field. Specifically it was designed to reproduce the "tail" bearing-journal design and conditions generally found in circulating water pumps. Proportions of the test set-up were derived from the dimensions of the standard type of p mps manufactured by the authors' company.

The tests of bearing-journal combinations have shown that the amount of wear and length of life of sleeve bearings can be substantially improved by the application of proper material combinations. In these tests the most commonly used bearing-journal combinations showed wear rates of 675 to 9462 $\times 10^{-7}$ in. per hr with failure in less than 500 hr. This is a relatively short life compared with some of the combinations developed for these tests which are still running after 6000 hr and have wear rates of less than 10 $\times 10^{-7}$ in. per hr. At this stage of the testing it is difficult to differentiate between the various types of carbide combinations; but as has been suggested in several other bearing studies and by our test results, the harder materials show better performance. It also has been suggested, but not conclusively proven in this test or other tests, that better performance is obtained when the same carbide material is used for both the bearing and journal. This is contrary to conventional practice with ordinary metallic materials.

The carbide combinations do not show signs of galling or seizure as experienced in the metallic combinations. The elimination of galling and equal distribution of the wear between the bearing and journal produces a bearing and journal with the most desirable characteristics for long life and low wear.

### Industrial

Leading off an Industrial Plant Safety session on Tuesday afternoon, **W. C. Kramer,** Commonwealth Edison Co., spoke on "A Power Plant Safety Program." Mr. Kramer stressed that he and his associates are not professional safety engineers but engineers and operators concerned primarily with the economical and reliable operation of a power plant.

State Line Station is a coal-fired steam power plant, located in Indiana on Lake Michigan at the Illinois-Indiana state line. It has a capacity of over 600 Mw in three units ranging in age from 31 years to 4 years. The number of employees has ranged between 180 and 425, with 375 representing a good average for the last 20 years. The employee safety program has resulted in twice attaining 1,000,000 man-hours without a disabling injury. As of now the record stands at four years or 3,000,000 man-hours with only one disabling injury.

Mr. Kramer described the two committees charged with the responsibility of executing the safety program.

### Fuels

"Fundamentals of Fuel Oil Combustion" was reported on by **D. S. Frank** of the Pure Oil Co. The author's first statement, "Fuel oil burning is still more of an art than a science," is all too true and he set about clearing up some of the mysteries of the art. On the subject of fuel oil specifications, Mr. Frank told us that there are a few important specifications of fuel oil which must be recognized. Quite often proper perspective of the relative value or interpretation of certain fuel oil specifications are not properly evaluated.

The National Bureau of Standards Specifications for No. 6 Fuel Oil provides little information. This specification states—"Oil for use in burners equipped with preheaters permitting a high viscosity fuel." Viscosity is stated as 45 Saybolt Furol seconds as a minimum and 300 seconds as a maximum. Flash point 150 F minimum and water and sediment as 2.00 per cent maximum. This is a sum of the water by distillation and sediment by extraction. Maximum sediment by extraction shall not exceed 0.50 per cent.

The most well-known properties of fuel oil are gravity, heating value, viscosity, flash point, pour point, sulfur, and ash content. Some attention is now being given to metals content of the fuel. All these tests are performed under controlled laboratory conditions. The relative importance of these properties depends upon the use of the fuel oil.

These properties vary depending upon the sources of crude and the refining process involved. Gravity generally varies from 5 to 15 API. However, fuel oils of both lower and higher gravity are marketed as No. 6. It is important only because it designates the heat content per pound or gallon. The Btu content per gallon is determined from the weight of the oil as denoted by the gravity and the heating value per pound. The lower the gravity, the higher the heat content per gallon and consequently lower cost per million Btu's. However, the full potential may often not be realized unless other factors influencing the end result are carefully engineered.

The minimum permissible flash point is usually regulated by federal, state or municipal laws and is based on accepted practice in handling and use.

The significance of pour point in a viscous oil is relatively unimportant, as the oil must be heated to insure proper handling.

Sulfur may or may not be important depending on the equipment in which the oil is being used. In some process work, low sulfur fuel may be essential. When used as a boiler fuel, the sulfur content must be evaluated against higher potential maintenance of economizers and air heaters. The effect of sulfur in conjunction with metals content and ash may influence superheater tube deposit in high superheat boilers.

Since the permissible viscosity of No. 6 fuel oil can range from a low of 45 sec to a high of 300 Saybolt Furol viscosity at 122 F, it is essential to know the slope or viscosity index. This is determined by obtaining an additional viscosity. On No. 6 oil this second determination is made at 210 F. The ASTM Standard Viscosity-Temperature Chart is so constructed that for any given petroleum oil, the viscosity-temperature points result in a straight line. The viscosity at any temperature can then be determined from this line.

An oil burner, Mr. Frank pointed out, has three functions to perform: (1) Split the fuel oil into a vast number of small drops—the smaller droplet size, the more efficient the burning. (2) Introduce and mix the air for combustion of the fuel oil. (3) Shape the flame to conform to the firebox in which the fuel is burned. These three requirements must be met regardless of the type of oil burner used.

Burners are classified according to the method used for securing atomization and fall in three general groups: (1) air atomization, either low or high pressure, (2) steam atomization, and (3) mechanical atomization. The subject of burners is amply covered in available literature. For large boilers, the trend is toward either high pressure mechanical atomizing or steam atomizing

burners. For small industrial boilers, a low pressure mechanical atomizing burner such as the spinning cup performs satisfactorily. In process furnaces, the type of product can influence the type of burner for the job, normally a low pressure air atomizing burner is used.

Each type of burner has its limitation of maximum operability of viscosity. All references to desirable viscosity are at the burner tip. Too often preheaters are located at some central point and the desired temperature may be obtained at the preheater outlet, but the oil may lose temperature enroute to the burner. This is quite true in industrial plants where oil lines may be long, improperly insulated, or the insulation has not been maintained.

**R. E. Kittoe,** Electronics Corp. of America, also spoke at the Tuesday afternoon Industrial Plant Safety session on "Combustion Safeguard Interlock Systems for Multiple Burner Boilers." His discussion was limited to boilers equipped with manually lighted burners not exceeding four per boiler and with fuel being gas or oil. This automatically placed an upper limit of boiler size of about 300,000 lb of steam per hour. The control systems covered were those which employed flame safeguards of the flame sensing type.

While boilers equipped to burn gas or oil alone were the subject of this paper, safeguard systems are being employed on pulverized coal fired boilers. Such control systems, generally speaking, are custom engineered to suit each application and were therefore beyond the scope of this paper.

It must be borne in mind that it is not the function of a combustion safety interlock system to replace boiler operators. Rather, the purpose of such control systems is to assist the indispensable, well trained operator in a twofold manner: (1) To assure that he follows a safe operating procedure in the lighting off of the burners; (2) to react instantaneously to hazardous situations where the fastest human reaction is too slow.

Mr. Kittoe referred to the J. B. Smith (Factory Mutual Engineering Div.) paper presented at the 1959 ASME Annual Meeting which set forth a detailed study of the causes of furnace explosions. He went on to say that both Factory Mutual Engineering Div. and Factory Insurance Association have indicated in their publications their recommendations for safe operation of multiple burner boilers based on their experience. Their recommendations on interlocks which parallel each other fairly closely was summarized by the author.

A sharply different set of conditions hold for the large boiler units. **Ludwig Skog, Jr.,** and **Edward H. Finch,** Sargent & Lundy, undertook to discuss "Some Aspects in the Control and Interlocking of a Gas-Fired Boiler-Turbine-Generator Unit" at an entirely different session. Wide variations exist between plants with respect to the control, interlocking, and protective schemes employed, Messrs. Skog and Finch noted, and, there are many reasons for these differences. For example, utility operating personnel have individual preferences; utility management have varying feelings on the degree of responsibility they are willing to place in the hands of an individual operator; manufacturers differ in opinion as to what constitutes adequate protection for major units of equipment and also individual plant design engineers approach the problem in sharply different ways.

At the present time, it appears that some degree of standardization has been reached in boiler combustion control systems, circulating water systems, turbine protective devices, and most of the elements of the boiler feedwater system. Still unresolved, however, are (1) boiler ignition and burner protective systems, (2) interrelation of boiler-turbine-generator interlocks and (3) protection of system against loss of a boiler feed pump. This paper presents, in the authors' opinion, a workable scheme for the control and interlocking in each of these controversial areas for large, modern, gas-fired, multiburner units of the type currently being installed by the electric utility industry.

Past practice has been built on the premise that no group of electrical or mechanical components were as reliable as the human eye, and no interlocking or protective system could be devised that could be substituted for a well-trained operator's judgment. This premise is still true, to a large extent, however it is open to serious question as to whether even the best operator's judgment and experience can be relied upon for the correct decision every time. However, it is felt that the most logical system to follow on boiler light-off is a semi-automatic scheme, in which the operator initiates each step of the procedure, preferably from some remote location such as the control room. In this system each step is completed automatically and its successful completion is monitored back to the operator, who then may manually initiate the next step. The sequence of the steps is made mandatory by suitable interlocks. An error by the operator, or the failure of any component trips out either the individual burner at fault, or the complete fuel supply and forces a fresh start. As experience is gained with this system, and components are improved from a reliability standpoint this system can be made more and more automatic. The ultimate goal could be the complete light-off of a boiler from a cold start with a single push button operation.

The fundamental design considerations for such a light-off system should include a purge cycle, the ignition and burner operation, the role and the number of boiler trip interlocks.

Recently a working group of the AIEE Power Generator Committee presented a paper No. 59-1149 entitled "Minimum Recommended Protection for Unit-Connected Steam Station." This paper considered by the authors to be excellent is said to coordinate the protection of the electrical end of the unit with that of the turbine and the boiler, so that trouble in any part of the unit causes partial or complete shutdown of the unit, depending on the type of trouble involved.

This paper reflects a current change in the basic philosophy of tripping by a large number of operating utilities. In the past it has been considered of prime importance to protect the power system against false tripping of a unit, which may occur whenever the number of tripping devices in any particular plant are multiplied. However, as more units are added to the system, and the cost of plant equipment, and the consequent cost of a failure to clear a genuine case of trouble are multiplied, there has been a strong tendency to regard the protection of the unit as of prime importance.

The authors show in tabular form, a basic scheme for

the application of these tripping interlocks. They point out that some variations in the details of application must be made to provide for differences in equipment as supplied by various manufacturers, however these details should be checked back to determine that they agree with the fundamental plan.

The interlocking scheme suggested by the authors was then traced through as to function.

**B. B. Bonadio,** Indiana-Kentucky Electric Corp., **A. S. Grimes,** American Electric Power Service Corp., and **H. I. Lamphier,** Ohio Valley Electric Corp., collaborated on the subject "Measurement of Density and Moisture in Large Coal Storage Piles." The problem of making a physical inventory of a coal storage pile has assumed increasing importance as the size of power generating stations and their annual coal consumption has increased. With annual coal consumptions ranging up to four million tons per plant, a small percentage error in weighing becomes a substantial quantity of coal which is reflected as a difference between the book and physical inventories.

The coal storage piles have kept pace with the growth of plant size. Piles containing in the order of one million tons are not uncommon. A nominal error in measuring can now represent a substantial quantity of coal for such piles.

The taking of an inventory of a coal storage pile involves three measurements: the total volume, the average in-place density and the average in-place moisture. Satisfactory procedures are available for determining the total volume. In-place densities can be measured at a given point with devices such as the California Sand Cone or the Washington Dens-O-Meter which are devices used in highway and air field construction. The use of these instruments in coal storage piles is limited usually to measurements near the surface. The measurement of in-place moisture content requires the taking of a physical sample for analysis and, usually, is limited to measurements near the surface.

Feeling that it should be possible to develop a more satisfactory inventory procedure, American Elec. Pwr. Svce. Corp., Ohio Valley Elec. Corp. and Indiana-Kentucky Elec. Corp. undertook a study and development program. Nuclear-Chicago Corp. had available instruments for density and moisture measurement which appeared to offer the possibility of improved techniques. Consequently, Nuclear-Chicago Corp. was invited to and did participate in the program.

OVEC's relationship with the U. S. Atomic Energy Commission in providing power for its Portsmouth Area brought about a close interest by the AEC in the early stages of the development program described in this article. The AEC is now sponsoring further studies for possible improvement in the methods described in the article. The initial step was to establish calibration curves for the measuring devices. This the authors did.

To establish a logical basis for sampling a large storage pile, a small portion of a storage pile was sampled extensively. Twenty-five access tubes were driven on 5-ft centers in a square grid arrangement. Density and moisture readings were taken at each 2-ft of elevation over the total depth of 18 ft. The data thus obtained was used to obtain estimates of the variance of the density and moisture.

The sampling procedure must be based on the number of access tubes to be driven and the number of readings to be taken in each tube. Since driving access tubes is the most expensive part of this method of taking an inventory, the sampling procedure should call for the minimum number of access tubes consistent with the desired accuracy.

Two successive samplings of a portion of one storage pile which was undisturbed between samplings has given average densities differing by only 0.15 lb per cu ft.

An attempt was made to use the moisture probe to determine the moisture content of the two storage piles. The question, of course, was whether or not the chemically-bound hydrogen content of the coal storage pile could be established with sufficient accuracy. Core samples were taken with a hollow casing driven into the pile. The casing was made up of two-foot sections. The casing could be driven from 2 to 6 ft before the friction built up to the limit of the driving equipment. The casing was then pulled, emptied, and then redriven an additional amount until the required depth was reached.

For one storage pile the moisture probe and the core samples gave similar values for the moisture content which were statistically equivalent. However, for the second pile the results were drastically different. This difference was greater than could be explained by any reasonable error in the value used for the chemically-bound hydrogen content.

Consequently, core sampling has been used for determining moisture contents on subsequent inventories.

A number of questions are still unanswered; particularly what causes different coals to have different calibration curves and what effects moisture content has on the density probe readings. The AEC-sponsored research program is aimed at answering these questions. In addition, considerable attention is being given to the problem of satisfactory use of the moisture probe.

### Large Steam Turbines

In keeping with the Conference's history of close attention to the utility industry's problems two sessions were held on large steam turbines.

**John R. Carlson,** Westinghouse Electric Corp., chose as his subject "The Economics of Very Large Turbine Generating Units." Mr. Carlson held his remarks to only the specific economies derived from the installation of large turbine-generator units (400, 600 and 800 Mw). Studies have been made of turbine-generator installations of 500 and 600 Mw capacity, with some consideration being given to even larger ratings. In these considerations, the questions most often raised are, "What is the advantage of a very large turbine-generator unit over two one-half size machines?" "Are there real economic considerations that justify the installation of very large turbine-generator units?" From the studies, the author's company recently made, the answer is "yes."

The assistance of the boiler manufacturers was enlisted, and they have given assurance that boilers can be built to match any current thinking in large turbine-generator sizes. This can be done economically and without encroaching on the availability.

However, there is more to the economic problem than just the turbine-generator and boiler components of a power plant. To assist in this study, the services of United Engineers and Constructors, Inc., were enlisted

to evaluate the cost of various large steam plants and determine the cost of generating electricity. This study included plants consisting of cross-compound turbines of 2-400, 1-600, and 1-800 Mw capacity. It also included two multi-unit plants of 2400 Mw capacity, one consisting of four 600 Mw turbine-generator units and the other three 800 Mw turbine-generator units.

In all cases the steam conditions selected were 2400 psig, 1050/1000 F, exhausting at 1.5 in. Hg Abs., with eight stages of feedwater heating. The boiler feed pumps were assumed to be driven by noncondensing extraction turbines, taking steam from the main unit.

The fuel for all plants was pulverized coal. Land, water supply, rail facilities, highways, and subsoil conditions were considered the same for all sizes of units.

The cost estimates which the author presented in a well detailed table are based upon prices for the major equipment and auxiliaries as of 1959. In the case of the 2400 Mw multi-unit power plants, the equipment was assumed to be erected consecutively and when allowances were made for certain items common to all units, some saving resulted over a single unit station.

From Mr. Carlson's tabulation the two major portions of the electrical energy costs proved to be the fixed charges and the fuel cost with the fixed charge of the greatest significance. Any reduction therefore in fixed charges obtained through size factors, design simplification or any other means in the author's opinion, should be, the most beneficial way of reducing the energy cost.

The energy cost of different sized steam plants was then plotted, all at a fuel cost of 25¢ per million Btu. Inasmuch as the heat rates of all three sizes of turbines ran about the same, the lower energy costs derived from the larger plants is obtained by the reduction in fixed charges due to the "size factor."

Further, a comparison of the energy costs of two 2400 Mw plants, one composed of 4-600 Mw and the other of 3-800 Mw turbines was shown. Here again the plant comprised of the largest generating units had the lower energy cost and this was held due to the reduced fixed charges.

Subcritical steam conditions, excepting for some pioneering, will be used for the majority of new plants in the immediate future, and the general availability should improve very materially and should approach that of the average operating conditions. This, Mr. Carlson believes, again merely proves that the price of progress is trouble, and the troubles are largely behind us.

In closing Mr. Carlson stated that experience with the large turbines of the kind discussed will pave the way for the next general advancement in steam conditions to 3500 psig and 1050 F, or higher which are desirable for these ratings.

The Philo, Eddystone and Avon turbines are already providing the operating experience necessary for the advanced steam conditions necessary for even larger turbines.

"Factors Influencing the Reliability of Large Steam Turbine-Generator Units" was presented by **C. D. Wilson,** Allis-Chalmers Mfg. Co., as a review of some of the basic principles that influence the reliability of large steam turbine-generator units. Design, production and installation considerations currently employed by the author's company plus examples of operating and maintenance practices that experience has shown to be most successful were used to illustrate these basic principles.

"Four Years Operating Experience with Large, Liquid-Cooled Turbine-Generators" was offered by **C. E. Kilbourne** and **C. H. Holley,** General Electric Co. The first large, liquid-cooled turbine-generator in the world was initially synchronized at the Eastlake Station of the Cleveland Electric Illuminating Co. on March 17, 1956. This generator, originally rated 260,000 kva, later, after confirming temperature tests, was rerated at 305,000 kva, is today accompanied by eleven sister units operated by eight different electric utility companies. These twelve generators, having a combined maximum capacity of over 3,000,000 kva, have accumulated over 19 machine years of operating history. This paper summarizes the experience of these years of machine service.

Now—how have the twelve operating units performed? To the best of their knowledge, there have been only two forced outages, one of 45 days, and one of 48 days. Both can be attributed to the "growing pains" of liquid cooling. In short, the availability of these units from this viewpoint has been 98.67 per cent. In addition to these forced outage days, these units have experienced planned outages whose total of 80 days is attributable predominantly to problems associated with the growth of generator capacity above 200,000 kva.

Starting with the very first machine, liquid-cooled units have performed well mechanically and thermally. Because of the design similarity to conventionally cooled units, there have been no new mechanical problems in the structural parts of liquid-cooled generators. Thermally, the machines have been outstanding in comparison with the conventional methods of cooling. It has been possible to increase the output of these generators substantially without increasing frame dimensions and weights, and still preserve conservative temperature rises. For example, the first installation at Eastlake was rated 260,000 kva at 30 lb hydrogen pressure, whereas the same frame, for a conventionally cooled machine, could at that time be rated only 162,000 kva. In spite of the added 100,000 kva output, a 62-per cent increase over the previously obtainable output, the temperature rise of the cooling oil passing through the stator winding was 38 C, with an expected copper hot spot temperature of approximately 42 C. This corresponds to allowable rises with the conventionally cooled machine of 60 C by RTD. The high-voltage bushings, which are arranged to use internal gas cooling, showed a rise of only 11 C when carrying rated current. With the gap-pickup rotor cooling, the rotor winding showed an average rise of 38 C over the cold gas temperature. These moderate rises, associated with the already realized substantial increase in output, are significant characteristics of liquid cooling.

The complete freedom from liquid leaks in the generator has been accomplished by rigid quality control tests that have been standard since the introduction of liquid cooling.

**Owen K. Brown,** Niagara Mohawk Power Corp., and **J. W. Batchelor,** Westinghouse Electric Corp., teamed up in the paper "Operating Experience with Inner-Cooled Turbine Generators." Inner-cooling has been applied successfully to fifty-five generators in operation today without any serious failures occurring in any component

of the ventilating system. Reports are included of operating experiences on these machines.

The first of these generators with inner-cooling on both rotor and stator was installed at Unit no. 66 in the Charles R. Huntley Station of the Niagara-Mohawk Power Corp. near Buffalo, N. Y. This generator was first synchronized with the system on Jan. 19, 1954, or more than six years ago. A report of the operation of this generator was included. The inner-cooled design is based on heat removal from the rotor and stator conductors by gas flowing within the confines of the ground wall of insulation. It should be noted that there is very little difference in the construction of the inner-cooled generators and preceding types with the exception of this ventilation pattern.

Application of inner-cooling to generator design served to reduce drastically the amount of material necessary for a given rating. For instance, the rotor weight was reduced to almost one half the weight with the conventional design. This meant, of course, that the excitation requirements would be increased substantially, but this could be provided for without exceeding the limits of experience already available. The service record of these machines has been excellent. There have been only a few cases of trouble involving inner-cooling directly, and none of these were of a serious nature.

With a change in relative dimensions of the inner-cooled rotor, it has become usual practice to operate the rotor above the second critical speed. With present day balancing techniques, this has not introduced any problems. On the other hand, it is found desirable to de-tune the shaft extension on three early machines, and make a closer check on the later machines before shipment.

"Operating Experience with Conductor-Cooled Turbine Generators" was the subject chosen by **L. M. Abrahamson,** Wisconsin Power and Light Co., and **L. T. Rosenberg,** Allis-Chalmers Mfg. Co. Rising fuel costs made the improved economy of the large steam turbine attractive. It was apparent that some means had to be found that would afford a substantial increase in the power output from the generators it was possible to build. Studies indicated that the most promising gains could be achieved by circulating an adequate quantity of the cooling medium within the main insulation wall.

This was accomplished on the first supercharged rotor by means of a powerful blower, and specially milled conductors, having a liberal area of contact with the high velocity hydrogen. Since the construction features of that design were previously presented to the American Power Conference, only a brief review was presented.

The initial installation with conductor-cooled rotor was subjected to wide daily load fluctuations ranging from 25 to 70 Mw, sometimes within an eight-hour interval. In order to minimize the effects of differential expansion on the conventional stator winding, a method of water control was introduced in June 1954. This consisted of adjusting the water flow to the coolers so as to maintain a generator coil temperature as constant as possible at all times. A remotely controlled valve in the inlet water header to the coolers was provided, with the controls located underneath a recorder which indicates water flow and coil temperature. On the subsequent larger unit at the Nelson Dewey Station automatic water control was planned.

A common misconception about conductor-cooled machines is that the high pressure blower causes excessive gas leakage. By comparison with the average static pressure within the casing, ranging from one to three atmospheres, the blower pressure is very low. The average gas leakage for the above three typical machines with conductor-cooled rotors for the past six months was 42.2 cu ft per day.

**J. E. Downs** and **E. H. Miller,** General Electric Co., discussed "Large Steam Turbine-Generators with Spinning Reserve Capacity." This paper describes a concept and suggests methods of designing a highly efficient steam power plant for base-load operation. Then, by drawing upon the characterstics which are designed into this station's equipment to obtain maximum efficiency and nearly continuous availability, low cost additional "spinning reserve" capacity can be provided for operation during periods of maximum system load. Thus, the features of two types of units are obtained in a single steam turbine-generator.

As the authors indicated, many of the components of a base-load station are designed quite liberally in order to provide the low velocities, the small pressure drops, and the good terminal temperature differences that are essential to obtain maximum efficiency in a base-load generating plant. Some examples of components sized primarily with these considerations are: turbine exhaust annulus; condenser and circulating water system; feedwater heaters; and the boiler air heater and economizer. Other components are sized primarily to give minimum maintenance and outages, assuming continuous full-load operation. For example, heat exchanger equipment is designed for low water velocities at maximum load to prevent erosion with continuous duty. Similarly, in a boiler designed for coal firing, liberal passages and volumes are provided to prevent excessive ash erosion and slag accumulation with the poorest fuel.

These factors, together, suggest that a steam power plant designed for base-load conditions could be substantially extended in capacity for short periods of time at some sacrifice in efficiency, with the expectation of some extra maintenance, and with perhaps only small increases in installed cost.

Increasing the initial pressure beyond the base-load design pressure appears to be one feasible method of exploiting the component design characteristics to obtain "spinning reserve" capacity. Approximately 30 to 40 per cent extra capacity may be obtained in this manner if the pressure is increased by a similar percentage. "Spinning reserve" capacity may also be obtained without increasing initial pressure if extra-high-pressure steam is generated but is bypassed around the turbine high-pressure section and is readmitted at a lower pressure to produce additional power. The extra steam bypassing the turbine high-pressure section amounts to approximately 84 per cent of the base-load throttle flow.

The incentives for reduction of initial temperature, and final feedwater temperature do not exist to the same degree as for the increased pressure scheme described earlier. Accordingly, studies to date of the steam bypass arrangement have been on the basis of maintaining full temperature at maximum load and with all feedwater heaters in service.

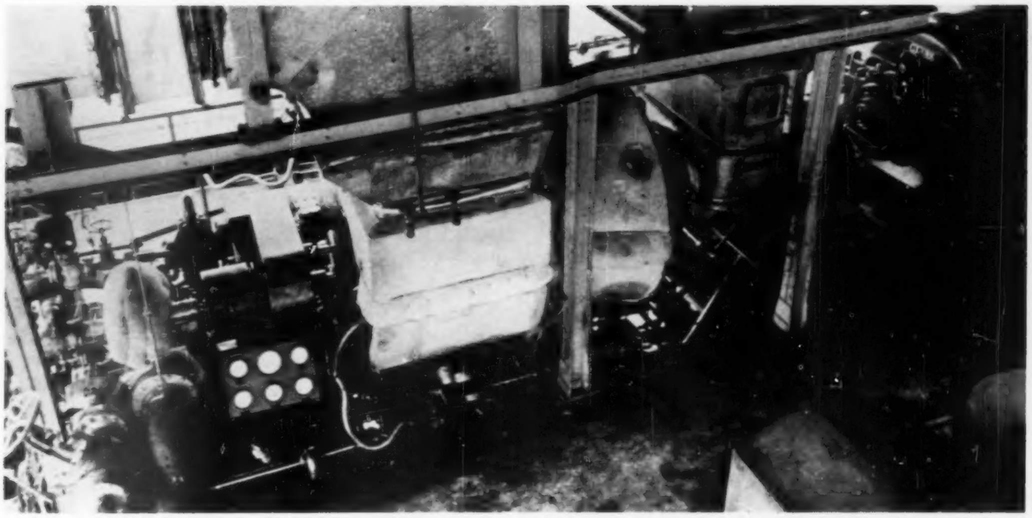

Fig. 1—View in Wakayama gas-turbine installation; 150-hp steam turbine for starting at left. Next in order are gas turbines, combustion chamber, axial compressor, and driven main air blower for fluid cat plant

# Development Experience With Prototype Gas Turbines*

By J. E. COOK†

Standard-Vacuum Oil Company

IN 1955 the Standard-Vacuum Oil Company embarked upon a program to modernize two refineries which included the installation of Esso Research Model IV fluid catalytic-cracking units with gas compressors and air blowers rated at 2300 and 2100 hp each, respectively. The catalytic-cracking plants were constructed in Wakayama, Japan, and in Palembang, Sumatra, and were of essentially the same capacity.

### Fluid-Catalytic-Cracking Plant

For the benefit of those who may be unfamiliar with petroleum refining, a brief description of a fluid-catalytic-cracking plant may be in order. The plant consists essentially of two large vessels, operating as an integral unit. One vessel is called the "reactor" and the second, the "regenerator." Heavy petroleum fractions such as gas-oil, together with a synthetic alumina-silica powder which acts as a catalyst are fed into the reactor to produce high-octane gasoline, liquid-petroleum gas, and other fuel products. The products from the reactor are separated by conventional fractionation.

As a result of the reaction, carbon is formed on the catalyst. Flowing much like a "fluid," the catalyst circulates continuously between the reactor and regenerator. In the regenerator, the hot catalyst is contacted with air which burns off the carbon, thus renewing the catalyst for continued use. The fluid-catalytic-cracking plant generates an excess of gas which is partially utilized as inexpensive fuel for the gas turbine. One gas turbine drives a gas compressor which handles the cracked product gas for further processing and the other gas turbine drives the main air blower which supplies air necessary to burn the carbon off the catalyst.

### Gas Turbines Selected

During the conception of the projects in Japan and Sumatra, consideration was given to the use of gas-turbine drivers to fill this application, but since a proved design in this capacity range had not as yet been manufactured, gas turbines were considered to be unavailable. However, at that time, one manufacturer was projecting the design of a gas turbine rated at 3000 hp which, although being larger than our apparent requirements, could be utilized. Since this machine was a new design and the manufacturer was rather anxious to put the unit into commercial service, Standard-Vacuum was made an attractive price for the purchase of four identical machines. The price was substantially a standoff when compared with full-condensing steam-turbine drivers together with associated cooling water and boiler

* Presented at the ASME Gas Turbine Power and Hydraulic Conference, Houston, Tex., March 6–9, 1960.
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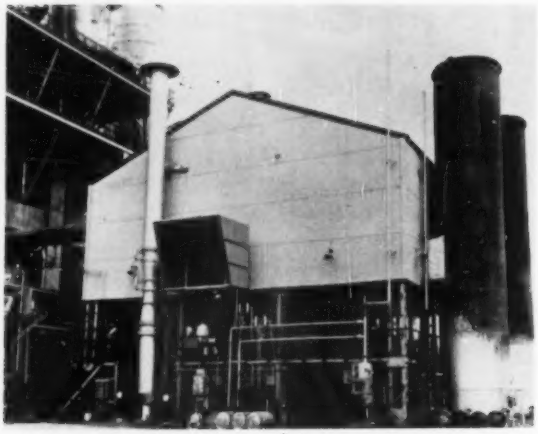


Fig. 2—View of gas turbine house, Wakayama Refinery. Proximity of short exhaust stacks (right) and air intake (center) tend to permit recycling of exhaust gas and increase intake-air temperature. Greater separation is desirable

capacity required for their operation. It should be emphasized, however, that the decision to install gas turbines was strongly influenced by the attractive fuel economy which would result through the burning of tail gas produced by the cracking unit. This decision was further influenced by the fact that these prime movers would be wholly controlled *on site* by the process operators as opposed to divided responsibility where utilities are generated at a remote powerhouse.

### Early Operation

The Wakayama machines went into service at the end of December 1956 and the Palembang machines were put in service September 1957. Considerable operating difficulty was experienced with the Japanese machines during the first year of operation. These machines were the first of a new line of gas turbines and naturally required some field development and modification of details subsequent to the startup. Since the Palembang machines followed the Japanese machines by some 9 months, it was possible to take advantage of the Wakayama experience and incorporate many of the desired changes in the Palembang machines before startup. As a result, relatively little operating difficulty has been experienced with the Palembang machines. The Palembang machines were not taken out of service for major inspection during the first 14 months of operation. The Wakayama machines were taken out of service in August 1957 after 7 months of service for a general inspection coincident with the cracking unit turnaround. It is the company's practice to review operating and maintenance experience gained during the first year of operation of prototype units.

### Difficulties Experienced

The difficulties experienced during this first year were:

1. Carbon buildup on gas nozzle.
2. Ambient temperature in excess of design temperature.
3. Adjustment of thermocouples in the flame pattern for optimum control temperature positioning.
4. Failure of the Kingsbury thrust bearing.
5. Axial-compressor-blade fouling.
6. Accidental plant shutdown through faulty instrumentation.
7. Range of recording temperature indicators on lubricating oil system.
8. High sulfur content of the fuel gas.

*Sulfur in Fuel Gas*

Gas turbine blades are manufactured from alloy material, the composition of which is known to the general public only as a trade name. Blades operate in a hot gas atmosphere of 1350 F at 8600 rpm. This condition plus various transient stresses causes blades to work very near, and occasionally beyond, their fatigue limits. Therefore, any impurities carried in the fuel stream which react with the blade metallurgy inevitably result in blade damage.

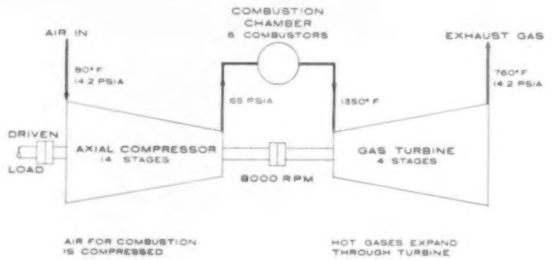



Fig. 3—Flow diagram for simple-cycle gas turbine showing pressure and temperature at significant points

The present machines were designed to burn a fuel gas that had a maximum of 3.6 per cent hydrogen sulfide content. The refinery was then running "sour" crude without desulfurization facilities with the consequence that the hydrogen sulfide in the fuel gas was normally about 4.5 per cent by weight. On infrequent occasions the hydrogen-sulfide content of the fuel gas exceeded 6–8 per cent for short intervals.

When the Japan machines were shut down for their first inspection after 7 months of continuous running, it was found that practically all of the blading in the first two stages, both rotating and stationary, had cracked about midway of the blade length on the trailing edge. As a result, all blading was removed, polished, and Prussian blued for careful examination. In this way hairline cracks were revealed in blades which previously appeared to be sound.

One complete spare rotating assembly and one complete set of stationary diaphragms purchased with the machines were installed in the air-blower driver thus restoring this machine to substantially new condition. After the repair of the first machine there was not a complete set of undamaged blades that could be salvaged to repair the second machine. As a result, one set of blades had to be reworked in the field to restore the second machine to service. Meanwhile, sample cracked blades returned to the United States for examination by independent metallurgical laboratories revealed the following:

1. Corrosion and cracking of the gas turbine blading was caused by high temperature sulfur corrosion.
2. To prevent similar failures in the future, only low sulfur fuel gas should be used. Indications are that the

Fig. 4—Cracked turbine rotor blade due to sulfur in fuel gas


Fig. 6—Cracked turbine diaphragm blades ground and drilled ready for reuse

sulfur content of combustion gas, after dilution with air, should not exceed about 40 grains/100 cu ft at temperatures between 1200–1400 F.

On the basis of this opinion, Wakayama installed desulfurization equipment to completely eliminate sulfur in turbine fuel gas.

Figs. 4 and 5 show the condition or rotating and stationary blades as found upon opening the turbine. Figs. 6 and 7 show the blades after reworking and ready for continued service. Reworking involved grinding away a large area of material surrounding the cracks and redressing the blade. A small hole was drilled at the end of some cracks in the stationary blades to serve as a stop against continued cracking. This is clearly shown in Fig. 6. After blade assembly, the rotors were rebalanced in one of the large industrial machine shops in Japan and returned to refinery service.

It is interesting to note that at the next general inspection 8 months later there was no evidence of further cracking of the reworked blades. It is equally interesting to note that hairline cracks were discovered in 13 of the new blades which had been installed earlier in spite of the fact that both machines had been burning sulfur-free fuel during these 8 months. This experience leads Standard-Vacuum to the conclusion that while sulfurous fuel may be a contributory factor, it is not the basic reason for the cracking of gas-turbine blades. This problem remains the subject of extensive research by all parties concerned.

The company's keen interest in this aspect of machine service life will be readily understood when it is known that the factory cost of a new rotor and a set of diaphragms has been quoted at $183,000. The deleterious effect of sulfur in fuel upon the life of gas turbine blading is well known throughout the industry. Our corrosion experience with an average of 1 to 3 per cent excess sulfur burning in a high temperature zone has been somewhat shocking—both to ourselves and to the machine designers.

In order to prevent turbine blade cracking in the future, we understand that the manufacturer is cutting $^3/_{16}$ in. from the trailing edge of the first two of the four stages, thus increasing the blade thickness at this point on all future blade deliveries for these machines. They advise that this modification will reduce the engine efficiency about $^1/_{10}$ per cent, but contend that reduced maintenance costs firmly justify this loss of engine

Fig. 5—Cracked turbine diaphragm blades due to sulfur in fuel gas

Fig. 7—Turbine rotor blades with cracks ground out ready for reuse

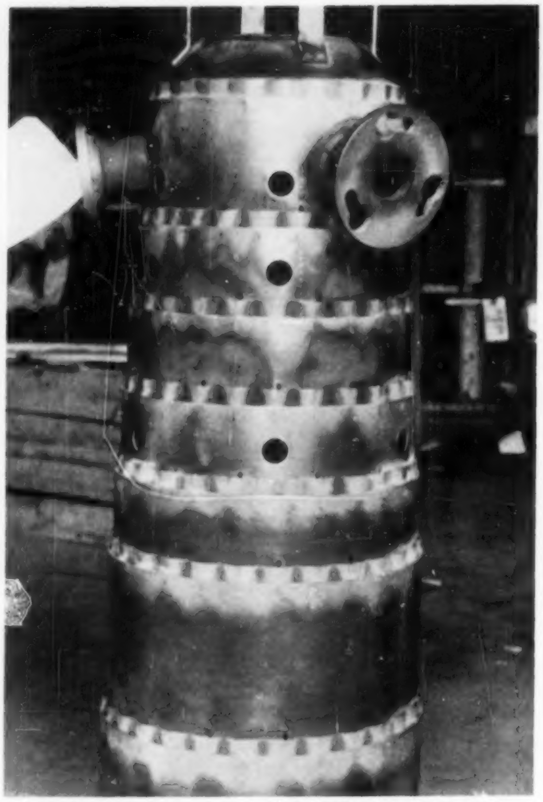


**Fig. 8—Combustor basket. Note small drill holes at each step to increase secondary air for improved combustion**

efficiency. Engine efficiency of these machines is normally about $17^1/_2$ per cent maximum.

*Carbon Buildup on Burner Nozzles*

During the several test runs before and after plant start-up, the operations were plagued with heavy deposits of carbon on the burner air baffles. Fig. 8 shows the combustion-basket assembly, and Fig. 9 the carbon formation on the baffle plate surrounding the burner nozzle. It was determined by laboratory test that carbon buildup resulted from burning fuel having appreciably higher heating value than that contemplated in the design specification. The condition was corrected by drilling additional holes in the combustor basket to increase the volume of primary combustion air. Carbon buildup also was experienced with the Palembang machines, but in this instance the remedy was to weld closed some of the holes in the baffle plates, thus reducing the amount of primary combustion air. Subsequently all combustors were redesigned and replaced by the manufacturer.

Experience indicates that combustor design is sensitive to the heating value of fuel burned. Obviously a great deal of trial and error was necessary at each of the two refineries before the optimum air/fuel ratio was established and burner cleanliness was obtained.

*Kingsbury Thrust Bearing Failure*

Within a few days after start-up following the August turnaround in Japan, the gas compressor driver dropped out of service under load. The Kingsbury thrust bearing failed for lack of lubrication. (It will be remembered that this is the machine which was fitted with the doctored cracked blades during the turnaround. But this fact has no bearing upon the thrust bearing failure.)

The lubrication failure is charged to two accounts: (1) The release of welding slag in the oil line partially closed a $^3/_8$ in. orifice, and (2) the improper application of temperature recorders to sense lubricating oil temperature.

Much of the 3 in. and 1 in. oil circulation system was of welded construction. The lines were picked, cleaned and oil flushed in the factory and oil flushed in the field before start-up and again during the turnaround. It is believed that a fragment of welding slag lodged in a recess in a flanged coupling, became dislodged as a result of vibration, and was carried by the oil stream to a $^3/_8$ in. orifice plate at the thrust bearing inlet nipple where it may have obstructed the oil flow to the thrust bearing. A partial rather than complete cut-off of lube oil was indicated by the temperature record. This showed a steadily rising oil temperature for about 15 min and was sustained for about 1 hour prior to failure.

The original installation included a single temperature printer-type recorder having a range of 0–1100 F and upon which nine temperature points were recorded. Five of these points recorded lubricating oil stream temperatures ranging between 105 and 160 F making temperature readings on the low side of the chart illegible. The operator had apparently neglected to observe the 16 deg F temperature rise produced by the failing thrust bearing. Only an alert and experienced operator, conscious of the implications of this particular temperature change, would have sensed the impending danger to the machine. This experience emphasizes the necessity for intensive operator training, especially on prototype equipment. It also emphasizes the necessity for providing a separate low range recorder for lubricating oil temperatures only. In addition to installing a 0–300 F recorder for this purpose, thermowells were installed to measure bearing-metal temperatures. These are recorded on the same chart with the oil streams.

The machine sustained severe internal damage to blading and shaft. The Kingsbury bearing was thoroughly fused, the shaft bent, and both the turbine and axial compressor blading scored and bent. Thrust is in the direction of the axial compressor so that the

**Fig. 9—Burner nozzle and primary air baffle plate, showing carbon build-up on baffle plate**

Fig. 10—Kingsbury thrust bearing collar showing effect of intense heat resulting from lubricating oil failure under load

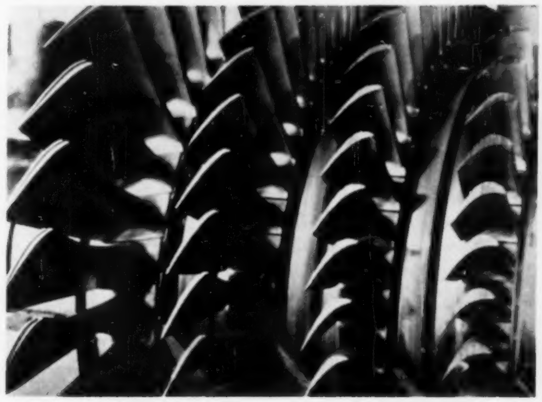


Fig. 12—Damaged turbine blading after lubricating oil failure

bearing failure permitted the rotor to slip 0.435 in. in the direction of the driven load, causing rotating blading to contact and deform diaphragm blading. Figs. 10 and 11 show the Kingsbury thrust-bearing collar and thrust bearing shoes after lubricating oil failure. Fig. 12 shows the damaged turbine blading after the failure.

The machine was repaired in 21 days by virtue of considerable improvising and the utilization of reworked blades, as well as replacement of the shaft by the one which had been removed from this machine during the August turnaround.

### Gas Turbine Performance

Standard-Vacuum has learned through experience that gas turbine performance is extremely sensitive to ambient temperature despite the fact that about 60 times the amount of air required for combustion is drawn through the machine. A power loss of about 1/2 per cent per degree F above 80 F can be expected when operating above design air-inlet temperature.

Performance is equally sensitive to axial-compressor fouling caused by the buildup of foreign material upon both the rotating and stationary blades. Load capacity drops from 3 to 5 per cent per pound drop in axial compressor discharge pressure. The combination of these two factors had reduced power availability from the design rating of 3000 hp to an actual 2100 hp before the significance of this sensitivity was realized. The condition was forcefully brought to attention when the Wakayama Refinery was forced to cut throughput for lack of gas compression power during a 100 deg F heat wave. The molecular weight of gas, at that time, was above the design specification which also increased the power demand. Except for the high ambient, plus axial compressor fouling, the gas turbine was adequate to meet load demand. Fig. 13 illustrates the effect of fouling upon power availability.

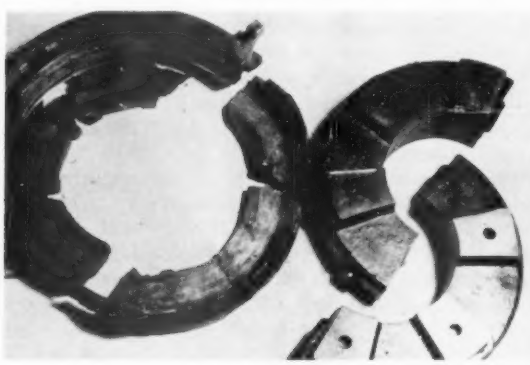


Fig. 11—Destruction of thrust bearing shoes resulting from lubricating oil failure

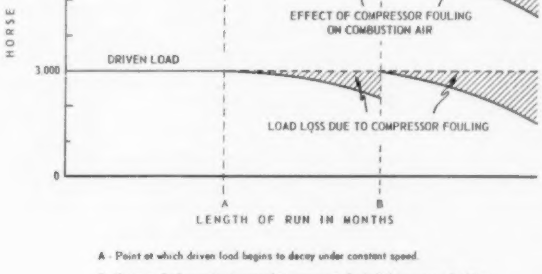



Fig. 13—Effect of axial compressor fouling upon power availability. Degree of fouling is indicated by reduction of compression ratio, and may be compensated for within design limits of equipment, by increasing engine speed to meet shaft horsepower requirements

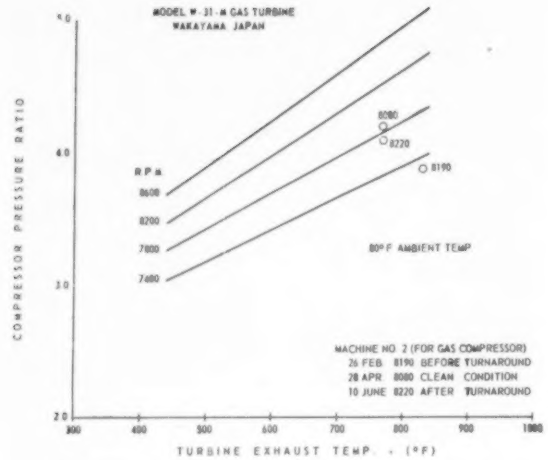



**Fig. 14—Standard compression ratio curves with field test points superimposed to indicate degree of axial compressor fouling**

It will be noted that approximately two thirds of the total internal power generated by the turbine wheels is consumed internally by the axial compressor, leaving about one third available for driving an external load. This accounts for the low thermal efficiency of simple cycle gas turbines.

The necessity for compressor blade cleaning without taking the machine out of service was obvious. Through the joint experimental effort of Esso Research and the manufacturer, a satisfactory method was devised. It was concluded that the scouring action of a soft, dry abrasive was required, and 200 mesh spent catalyst was found to be effective.

The degree of fouling can be determined by noting the loss of discharge pressure in the axial compressor as indicated in Fig. 14. A power output loss of about 4 per cent per lb pressure drop can be expected. When axial compressor outlet pressure drops from the normal of 50 psig to about 46 psig, representing a shaft output power loss of about 500 hp, catalyst is hand fed into the air-intake stream at the rate of 3 lb per min for 15 min. Experience shows that substantially complete cleaning is effected within the first 5 minutes of the injection of spent catalyst.

Figs. 15 and 16 were plotted showing the spent catalyst injection rate per unit time versus the operating characteristics of the turbine. The efficiencies shown are based on operating instrumentation and are, of course, only relative indicator for operation.

Inlet-air temperature can be lowered by the installation of air washers. Here extreme caution is necessary to insure effective water baffling. The use of treated boiler water is essential. It is reported that the use of untreated water and inadequate baffling in one powerhouse installation in South America accelerated compressor fouling beyond the calculated benefits of lowered air-inlet temperature. This condition has been corrected satisfactorily.

### Conclusions

It is emphasized that a willingness to accept prototype equipment axiomatically implies a willingness to accept joint responsibility for the correction of engineering details associated with new design concepts, unpredictable plant interruptions and downtime, loss of throughput and much mental anguish on the part of both operators and management. There were occasions during the first year's operating experience when Standard-Vacuum felt that the sponsoring of prototype machines had been a questionable decision. But with this experience behind us and with a fuller understanding of the capability and limitations of the machines, the company is looking forward to an extended period of reliable and economic power for cat-plant air blowers and gas compressors with gas-fired-turbine drivers of this capacity. The economic utilization of waste-fuel products is a strong incentive for pioneering in the application of gas turbines.

We feel rather strongly that gas turbines should be designed for about 15 to 20 per cent excess capacity over maximum load requirements. This would permit a reasonable amount of compressor fouling and seasonal rise in ambient without the necessity for reducing load on the driven unit. This added design capacity should be accepted as a fouling factor just as is done in the case of heat-exchanger design to afford more realistic evaluations of gas turbine performance under actual operating conditions.

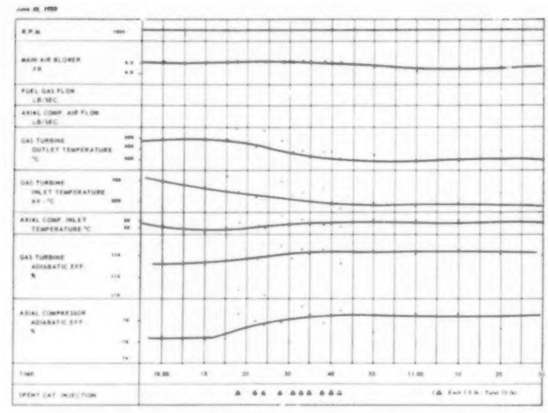


**Fig. 15—Air-blower-turbine, spent-catalyst cleaning data**

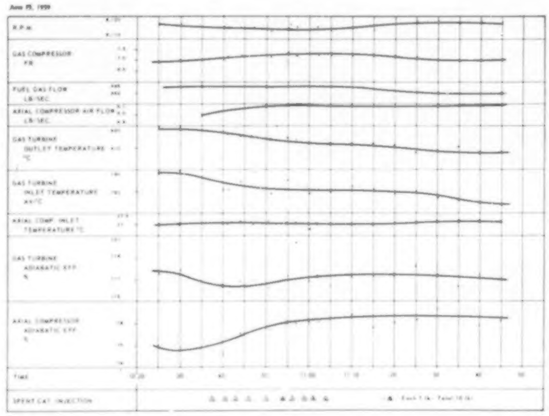


**Fig. 16—Gas-compressor-turbine, spent-catalyst cleaning data**

# Abstracts From the Technical Press—Abroad and Domestic

**(Drawn from the Monthly Technical Bulletin, International Combustion, Ltd., London, W. C. 1)**

## Fuels: Sources, Properties and Preparation

**The Chemistry of Carbonization Investigated on Polymeric Bituminous Coal Model Substances VII. Application of the New Insight to Bituminous Coal.** D. W. van Krevelen, P. M. J. Wolfs and H. I. Waterman. *Brennst Chem.* 1959, **40** (Dec.), 371–7 (in German).

The new knowledge gained from the results of the studies described in the previous articles is discussed with reference to its application to the coking mechanism of bituminous coal. Analogies have been found for the temperature range of decomposition and plasticity, selectivity of gas and tar formation, course of composition of decomposition products and influence of temperature-time history. A new equation for the condensed ring number has been derived and the average number of bridges per average structure unit is probably 1–1.3 for bituminous coal. Tentative conclusions are drawn of the functional distribution of the elements, especially hydrogen.

**Thermo-gravimetric Studies of Coal and Model-substances.** J. A. Corrales and D. W. van Krevelen. *J. Inst. Fuel* 1960, **33** (Jan.), 10–6.

The investigations included: (1) physical devolatilization and chemical decomposition of pure substances; (2) vaporization of mixtures; (3) mixed vaporization and decomposition; (4) vaporization of substances absorbed inside porous materials; (5) decomposition of coal; (6) devolatilization of impregnated coals.

**Prediction of the Coke Quality from Composition of Coking Blends of Medium- and Low-volatile Coals.** D. W. van Krevelen, F. J. Huntjens and A. H. Wilms. *J. Inst. Fuel* 1960, **33** (Jan.), 3–9.

It is shown that results obtained by the dilatometer test can be used to decide to what particle size the coals in a blend have to be ground to produce a coke of satisfactory quality.

## Steam Generation and Power Production

**Thermo-dynamic Properties of Natural and Heavy Water.** V. A. Kirillin and S. A. Oulibin. *Teploenergetika* 1959 (Dec.), 77–80 (in Russian).

Experiments have been carried out into the density of heavy water. The thermal properties of $D_2O$ and $H_2O$ are compared.

**Reference Values of Specific Heat $c_p$ for Water and Steam.** A. E. Sheindlin, E. E. Shpilrain and V. V. Sichev. *Teploenergetika* 1959 (Dec.), 80–3 (in Russian).

Existing experimental values for $c_p$ for water and steam are analyzed and a table of reference values has been compiled for pressures of up to 700 kg/cm.² and temperatures up to 650 C.

**Reduced Coefficients of Frictional Resistance during Flow of a Steam/Water Mixture in Pipes.** N. I. Semenov and B. I. Sheinin. *Teploenergetika* 1960, **7** (Jan.), 33–7 (in Russian).

Test data on the resistance of slightly inclined pipes during the flow through them of a steam/water mixture are analyzed and compared with such information in regard to an air/water mixture.

*C. E. G. B. abstract*

**Pressure Vessel Overtemperature Hazards.** D. B. Rossheim, J. J. Murphy, G. P. Eschenbrenner and R. S. Eagle. *A.S.M.E. Preprint* No. 59-A-319, 1959, (Dec.), 13 pp.

The nature, sources, effects and means of detection and control of excess temperatures in pressure vessels are considered and suggestions made for design, operating precautions and establishment of a code.

**Thermodynamic Investigations of Boiler Heating Surfaces at High Heat Release Rates.** K. R. Schmidt. *Mitt. V.G.B.* No. 63, 1959 (Dec.), 391–401 (in German).

The occurrence of film evaporation, i.e., the formation of a steam cushion of low thermal conductivity on the inside of tube walls, has been investigated as a function of flow velocity, tube diameter, direction of flow, wall temperature, pressure and heat release rate. The results are presented in graphs and discussed.

**Problems of Forced Flow Boilers with Special Attention to the Steam and Water Flow.** R. Michel. *Mitt. V.G.B.* No. 63, 1959 (Dec.), 402–14 (in German).

The arrangement of the various heating surfaces in a forced flow boiler, i.e., evaporation tubes, economizer, superheater, reheater, is discussed so that everywhere sufficient flow velocities are obtained, excessive temperatures prevented and the risk of film evaporation is avoided. Closed starting and stopping circuits are recommended.

**Recent Boiler Designs with High and Ultra-High Steam Parameters.** K. F. Roddatis. *Teploenergetika* 1960, **7**, (Jan.), 3–13 (in Russian).

A review is presented of the larger Russian-produced boilers and their elements based on materials supplied by the Ordzhonikidze Krasnyi Kotelschik Factory, and of decisions of technical committees of the Rostov and Moscow Councils and Ministry of Power Station Construction. Details of numerous boilers are set out in 7 tables for comparison.

*C.E.G.B. abstract*

**An investigation into the Performance of a TP-170 Natural Gas Fired Boiler with Variable Feed Water Temperature.** G. M. Poliakov, A. V. Ilyin, A. V. Zimachinsky and G. A. Schapov. *Teploenergetika* 1959 (Dec.), 51–5 (in Russian).

The results of tests are presented of operating the boiler at loads from 110 to 190 t/h and feedwater temperatures from 150 to 215 C. The relation of heat loss in the flue gas to the feed water temperature at constant evaporation is shown.

**Industrial Boiler Design.** T. B. Hurst and C. C. Hamilton. *A.S.M.E. Preprint* No. 59-A-78, 1959 (Dec.), 7 pp.

The design of boilers with gas, oil or coal firing is discussed and figures presented of heat liberation (Btu/cu ft/hr), heat release rates (Btu/sq ft/hr), relationship between furnace depth and heat release rate per ft burner wall, heat release rates for various types of grates (Btu/sq ft grate/hr), superheater tube spacing and range of steam mass flow values. Designs of furnace walls for pressure-fired and stoker-fired boilers are shown.

**Modern Steam Generators for the Firing of Blast Furnace Gas and Pulverized Coal.** K. Nuber. *B.W.K.* 1959, **11** (Dec.), 566–9 (in German).

The difficulties of designing steam generators for the firing of blast furnace gas and pulverized coal either singly or in any proportion are discussed and two examples shown of how these have been solved. In one plant with 3 boilers each rated at 110 klb/h, 1200 psi and 975 F the gas burners are installed in the rear wall of the furnace; to improve the mixing of air and gas there are five rows of six burners each with inserts in each

burner mouth to "tear up" the gas stream as it is mixed with the air; the coal is pulverized in a central plant, stored in intermediary bunkers and injected through corner burners. In the other plant with 2 boilers each rated at 90 klb/h, 1000 psi and 950 F gas burners of similar design are installed below the p.f. burners in each corner. Since the fly ash from these boilers contains up to 40 per cent combustibles there are added to each boiler small vertical water-cooled slagging furnaces in which a mixture of pulverized coal and fly ash is burned, the combustion gases being added to those in the main furnace and the water-cooling system forming part of the main circulation system.

**Inspection of Large Boilers.** G. Barnard. *Elect. Rev.* 1960, **166** (Jan. 1), 8–11.

It is suggested that in addition to the generally very careful inspections during fabrication of parts, erection on site and annual statutory overhaul, inspections of all electrical equipment and controls, steam lines, feedwater, tube samples and instruments be made by an independent authority to ensure trouble-free operation during the following operating period.

**Boiler Furnace Fuel Explosion Survey.** J. B. Smith. *A.S.M.E. Preprint* No. 59-A-303, 1959 (Dec.), 4 pp.

The causes of boiler explosions, especially of pulverized coal fired boilers, have been analyzed and the most frequent types detailed. Measures to prevent such accidents are discussed.

**Some Observations on the Specific Surface of Coals.** S. J. Gregg and M. I. Pope. *Fuel* 1959, **38** (Oct.), 501–5.

The specific surface of vitrains has been determined by nitrogen and n-butane adsorption at 273 K. The specific surface calculated by n-butane adsorption agrees with the heat of wetting values using methanol as wetting agent.

**Chemical Structure and Properties of Coal XXIV—Sound Velocity and Fraction of Aromatic Carbon.** D. W. van Krevelen, H. A. G. Chermin and J. Schuyer. *Fuel* 1959, **38** (Oct.), 483–8.

A method for calculating the fraction of aromatic carbon in coal from the velocity of sound, density and elementary analysis is described. The results obtained by this method are practically equal to those from the densimetric method.

**Contribution to the Research into Coal Structure by van Krevelen's Method.** V. Rona. *Brennst.-Chem.* 1959, **40** (Nov.), 346–54 (in German).

The calculation of the volumina of atoms, the accuracy of the determination of aromaticity and the theoretical basis and extension of van Krevelen's method are discussed. The results of these considerations are claimed to confirm that bituminous coal consists of individual molecules of average molecular weight and is not a polymeric substance built up of structural units.

**The Chemistry of Carbonization Investigated on Polymeric Bituminous Coal Model Substances. VI. The Plastic Behavior during Carbonization.** P. M. J. Wolfs, H. I. Waterman and D. W. van Krevelen. *Brennst.-Chem.* 1959, **40** (Nov.), 342–6 (in German).

Two extreme processes are shown to be possible during carbonization of these model substances analogous to the behavior of different coals and their plastic behavior can be explained simply by the position of the decomposition curve relative to the softening curve.

### Mechanical Handling

**Coal Bunker Explosions—Cause and Prevention.** W. W. Hagnauer and K. W. Hamming. *Coal Utiliz.* 1959, **13** (Nov.), 24–6.

The conditions in which gas may be evolved in coal stored in bunkers are discussed and the means described by which accumulations of gas and the possibility of explosions can be avoided. These include: (1) Sufficient ventilation to remove explosive gases; (2) elimination of coal dust in conveyor rooms by proper ventilation; (3) improved design of bunkers to reduce hang-up of coal with an illustration of preferred bunker design; (4) operating factors including regular cleaning and inspection.

### Furnaces and Combustion

**The Prevention of Acidic Smut Emission from Oil-Fired Boilers.** Anon. *Steam Engr.* 1960, **29** (Jan.), 114–5, 122.

Acidic smut emission from oil-fired Supereconomic boilers has been stopped by the injection into the flue gas of a finely divided alkaline hygroscopic powder (D.C.P.58) the particles of which provide nuclei for the absorption of acid and greasy particles. The agglomerated particles are subsequently extracted in a specially designed "Turbocell" dust collector. The design of the injecting device is described briefly. The introduction of this powder resulted in a reduction of the acid dew point of the flue gas by 20 F.

**An Experimental Investigation of Fuel Additives in a Supercharged Boiler.** R. J. Zoschak and R. W. Bryers. *A.S.M.E. Preprint* No. 59-A-160, 1959 (Dec.), 10 pp.

Pilot plant experiments are described in which various magnesium and aluminum compounds were used as additives to washed fuel oil and ash accumulations and corrosion studied. The ash deposits were generally very heavy but could be removed by soot blowing. The inhibited ash was not corrosive at temperatures up to 1600 F with an additive ratio of 1.5 Mg: IV. Tentative suggestions are made regarding the process of ash deposit formation.

### Steam Generation and Power Production

**The Critical and Two-Phase Flow of Steam.** W. G. Steltz. *ASME Preprint* No. 59-A-223, 1959 (Dec.), 8 pp. Analytical and digital-computer studies have been made and are compared to the critical flow of steam. Choking velocity, critical pressure ratio and critical flow per unit area are presented as a function of inlet conditions. A Mollier diagram gives the exponent in the wet and superheated regions.

**The Effect of Non-Uniform Heat Distribution on Steam Generation in Tubes.** P. G. Morgan. *Engng. Boil. Ho. Rev.* 1959, **74** (Nov.), 336–8.

A summary is presented of a Russian paper dealing with the effect of non-uniform heating on the transition from bubble to film boiling. An equation for the calculation of the critical heat flow has been developed.

**Development of Boiler Design by Sulzer Brothers Limited.** Anon. *Engng. Boil. Ho. Rev.* 1959, **74** (Nov.), 340–4.

A review of the boilers built by this firm and particularly of the monotube boiler and its controls.

**Steam Generators for Indirect Steam Generation.** O. H. Hartmann. *Energie* 1959, **11** (Nov.), 546–8 (in German).

The inventor of this type of boiler surveys the development and fields of application which are mainly small and medium industrial plants. The great advantage of this boiler is that the steam generated directly flows in a closed circuit which has to be filled once only with demineralized water when starting up and requires very little make-up, while the steam for consumption is generated in a drum by heat exchange with the directly generated steam and the feedwater requirements are thus less onorous. Various designs of boilers are illustrated.

**Small Boilers for High Heating Temperatures.** K. Perl. *Energie* 1959, **11** (Nov.), 548–9 (in German).

The corner-tube boiler has proved particularly suitable for the generation of 100,000–200,000 kcal/h either at high pressures (up to 100 atü) when using steam as heat carrier or at low pressures (up to 6 atü) using a liquid heat transfer medium. Designs proved successful in operation are illustrated.

**Automatic Coal-Fired Boiler Plant.** Anon. *Steam Engr.* 1959, **29** (Dec.), 75–9.

A summary is presented of the papers to the conference held by the Combustion Engineering Assoc. in November 1959 dealing with various aspects of mechanical handling of fuel and ash, mechanical firing of boilers and automatic controls for smaller and larger plants. A few case histories showed the improvements obtainable by such controls.

**Vibration Cleaning Boiler Heating Surfaces.** V. I. Chastoukhan and E. L. Zarechansky. COMBUSTION 1959, **31** (Nov.), 39–40.

A system of vibrating the tubes of a La Mont waste heat boiler to free them from deposits is described. Vibration periods of $1^1/_2$–2 min. every 15–20 min. at a frequency of 28–30 c/s have proved successful.

**Belongs the Future to the Gasifying Burner?** F. Wilkens. *Energie* 1959, **11** (Nov.), 552–3 (in German).

The "Intherma" burner has been designed for use with Bunker C oil preheated to 160 F and gasified with superheated steam at 1020 F to produce a gas consisting mainly of methane. The resulting flame is soft and hot (1600 F) and because of very low excess air requirements the $CO_2$ content of the flue gas is about 15 per cent. Absence of corrosion in the low-temperature surfaces despite an exit flue gas temperature of 265 F

is ascribed to the fact that in the production of the methane all the hydrogen in the steam, oil and air is consumed so that any $SO_3$ formed cannot combine with hydrogen to form $H_2SO_4$.

### Water-Side Corrosion and Water Treatment

**Present-Day Feedwater Treatment for High-Pressure Boilers.** P. Hamer. *Engng. Boil. Ho. Rev.* 1959, **74** (Dec.), 368–71, 374.

An abridged version of the author's paper to the I.M.E. is presented which discussed: (1) Control of steam purity; (2) deaeration of boiler feedwater; (3) solids in steam and deposits; (4) steam washers; (5) control of corrosion in boilers; (6) corrosion in the absence of oxygen; (7) idle boilers; (8) acid cleaning.

**The Water in Power Station Operation.** P. Pracht. *Energie* 1959, **11** (Nov.), 514–26 (in German).

A review dealing with: (1) Components of boiler water; (2) acid cleaning of boilers on site or of boiler parts in the works; (3) feedwater requirements; (4) steam-water separation in the drum; (5) feedwater treatment methods; (6) feedwater treatment in industrial power stations; (7) cooling water treatment; (8) additions to the feedwater.

**Hide-Out Effect and Decomposition Phenomena in Heating Surfaces of Steam Generators at High Heat Release Rates.** E. A. Ulrich. *Tech. Überw.* 1959, **11** (Nov.), 395–8 (in German).

The occurrence of hide-out and film evaporation, their dependence on heat release rate, mass flow and salt content of the boiler water, and their effect on scale formation and tube failure are discussed.

**Solubility of Ferric and Cobalt Oxides in Dilute Hydrazine.** R. S. Young. *Ind. Chem.* 1959, **35** (Nov.), 549.

Experiments have shown that the protection against corrosion of a standby boiler filled with water containing hydrazine at normal temperature and pressure is due to direct reaction of hydrazine with dissolved oxygen rather than with ferric oxide.

**Factors Determining the Purity of Completely Demineralized Water.** A. Richter. *Tech. Mitt.* 1959, **52** (Nov.), 409–12 (in German).

The following factors are considered: (1) genuine ion exchange; (2) ion exchange with subsequent reactions; (3) adsorption processes;

(4) change of charge; (5) chromatographic effects. Their influence on the final quality of the water is discussed.

**Fundamentals of Complete Demineralization Plant Circuits.** H. List and J. Tregl. *Tech. Mitt.* 1959, **52** (Nov.), 417–23 (in German).

Arrangements of complete demineralization plants for obtaining and maintaining maximum efficiency and economy are discussed and illustrated by some actual examples.

**Semi- or Fully-Automatic Demineralization Plants.** H. V. Nodel. *Tech. Mitt.* 1959, **52** (Nov.), 428–32 (in German).

Both semi- and fully-automatic plants are described and the difference in their operation outlined. It is argued that although the capital costs of a semi-automatic control are 9 per cent and that of a fully-automatic control system are 28 per cent higher than a manually controlled plant it is almost always economically justifiable to install an automatic control system.

### Flue Gas, Ash and Dust

**The Residual Oil Ash Corrosion Problem.** C. J. Sunder. *Corrosion* 1959, **15** (Nov.), 61–6.

This review is mainly concerned with the problems of high temperature corrosion and their prevention. It is believed that steels containing silicon and aluminum may be developed and/or coatings resistant to corrosion temperatures above 1200 F. Oil additives giving equal protection for all variations of impurities, operating conditions and materials are less likely to become available.

**Control of Air Pollution from Oil-Burning Power Plants.** H. C. Austin and W. L. Chadwick. *ASME Preprint* No. 59-A-71, 1959 (Dec.), 4 pp.
Pilot and full-scale plant tests are reported on reducing the amount of particulate matter, $SO_2$, $SO_3$ and NO in the flue gases emitted from oil-fired boilers. Particulate matter and $SO_3$ can be separately removed in newly designed precipitators, the $SO_3$ also by adding a stoichiometrical amount of ammonia. $SO_2$ can be removed by oxidation over a vanadium catalyst; this method is very expensive. For the separation of NO, two-stage combustion has proved best and is at present under extended investigation in a large boiler. An infra-red gas analyzer has been developed for continuous recording of $SO_2$, $SO_3$ and NO which is much more accurate and reliable than previous instruments.

**The Use of Transparent Scale Models in the Design of Dust-Collector and Gas Duct Systems for Coal-Burning Electric Generating Stations.** E. F. Wolf, H. L. von Hohenleiten and M. B. Gordon. *A.S.M.E. Preprint* No. 59-A-305, 1959 (Dec.), 5 pp.

Improved dust separation by changes in design of the gas ducts to and from the collectors have been secured by means of transparent models. The difficulties experienced in a particular plant and the modifications introduced after model studies are described in some detail.

**A Superior Collecting Plate for Electrostatic Precipitators.** H. J. White and W. A. Baxter. *A.S.M.E. Preprint* No. 59-A-279, 1959 (Dec.), 7 pp.

Experimental studies have shown that the performance of a solid plate as collecting electrode is far superior to an expanded or perforated metal plate. The design of the plate with its baffles is described.

**Pulverized Fuel Ash as a Concrete Additive.** R. Hammond. *Mach. Lloyd* (Europ. Edit.) 1960, **32** (Jan. 9), 19–23.

The advantages and disadvantages of using fly ash in concrete construction are discussed and some applications in heavy foundation work and large dams described. Fly ash in concrete is particularly useful where its resistance to aggressive water can be utilized and early strength is of minor importance.

**The Use of Fly Ash in Concrete by Ontario Hydro.** J. N. Mustard and C. Macinnis. *Engng J.* 1959, **42** (Dec.), 74–79.

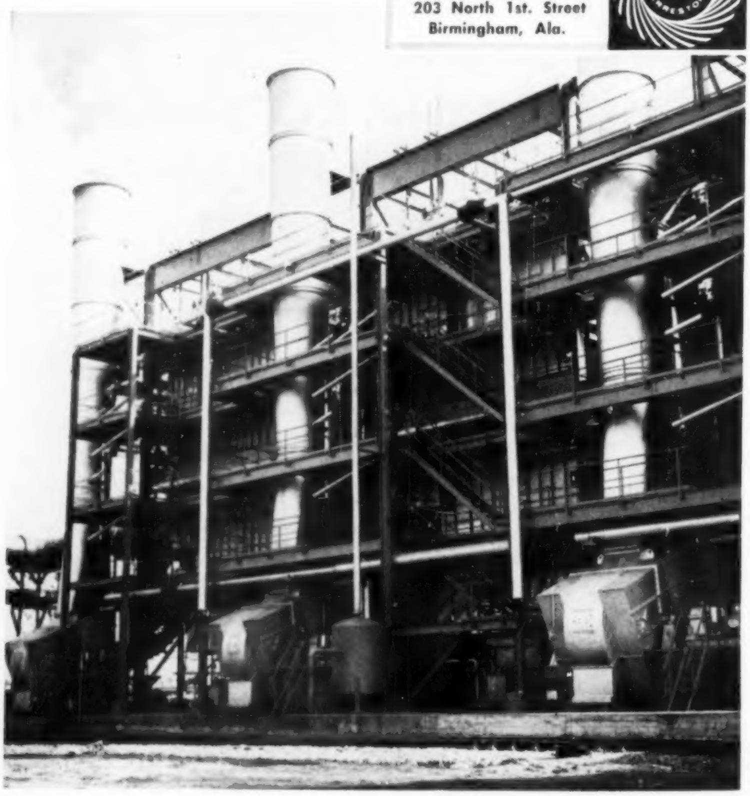
The properties of fly ash concrete are discussed and the use of fly ash in both high- and low-pressure grouting mixtures at Niagara and to replace from 20–30 per cent of the cement in appropriate mass and structural concrete applications at the St. Lawrence, Carribau, Whitedog and Otter Rapids projects are described. An account is given of the method developed for batching the ash as a slurry.

*From C.E.G.B. Digest* 1960, **12** (Jan. 30), 286.

### Heat Recovery Plant

**Dynamics of Heat Exchangers and their Models.** H. Thal-Larsen. *A.S.M.E. Preprint* No. 59-A-117, 1959 (Dec.), 12 pp.

Simple mathematical models are used to analyze the dynamic characteristics of heat exchangers and parameters, such as dwell time of the fluid in the heat exchanger and the metal and fluid time constants, are established to describe these characteris-



ties. A single dynamic parameter is proposed which is easily adjusted to different operating conditions and believed to be useful in the design, use, classification and standardization of heat exchangers.

**Systematic Approach to the Design of Compact Heat Exchangers.** W. B. Hendry. *A.S.M.E. Preprint* No. 59-A-193, 1959 (Dec.), 7 pp.

Equations are developed for designing a direct-transfer, cross-flow, single pass heat exchanger of minimum volume or minimum weight.

**Design Performance of The Low-Level Economizer.** J. H. Potter and R. C. King. *A.S.M.E. Preprint* No. 59-A-222, 1959 (Dec.), 12 pp.

The low-level economizer is installed behind the air preheater to remove further heat from the flue gases. Performance studies are presented which show that its installation is an economic proposition.

**The Low-Level Economizer.** S. Jewson. *A.S.M.E. Preprint* No. 59-A-226, 1959 (Dec.), 8 pp.

The design of the economizer, installed behind the air preheater and used to preheat feedwater, control of surface temperature, use of corrosion-resistant cast iron, and special washing equipment for the cleaning of the low-temperature heating surfaces are described.

### Power Generation and Power Plant

**Economic Selection of Steam and Power Supplies for Refineries and Petrochemical Plants.** D. L. E. Jacobs and W. B. Wilson. *A.S.M.E. Preprint* No. 59-A-330, 1959 (Dec.), 12 pp.

A general consideration is presented of the factors involved in selecting boiler size, pressure and temperature, choosing between extraction-back-pressure or condensing turbine and pressure reducing stations, between turbine and motor drives, etc. Two case studies demonstrate how to arrive at the most economic solution in a given case.

**Steam Power Plants for Peaking Service.** A. R. Lebailly. *ASME Preprint* No. 59-PWR-8, 1959 (Oct.), 6 pp.

The calculations presented show that considerable reductions in investment can be achieved if units for supplying peak load (maximum 1500 h/year) are designed for lower pressures and temperatures, higher furnace heat release rates and simplified accessories. Based on a 100 Mw unit, conditions compared are 850 psi and 900 F, 1250 psi and 950 F, 1800 psi and 1000/1000 F and 1450 psi.

# REPUBLIC VECTOR CONTROLLERS FEATURE 2 TO 500% PROPORTIONAL BAND ADJUSTMENT

Rockwell-Built Republic vector controllers feature easy proportional band adjustment from 2 to 500%, an adjustable reset rate from 0.1 to 50 repeats per minute and less than 0.05% dead band.

This is typical of the performance of Republic's new Null-Balance-Vector line of pneumatic instruments which includes:

- Temperature transmitters with a 10 to 1 range adjustment.
- Pressure transmitters of 0.5% accuracy, 0.1% sensitivity.
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- Relays which extract square root down to 10% of output span.
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- Other instruments necessary for measuring, controlling, and computing variables for combustion and process industries.

The span ("K") of each Republic Null-Balance-Vector instrument can be easily varied by loosening the lockscrew and sliding the weighbeam assembly along the slot as shown in the schematic. Zero does not have to be reset and only a comparison reference is required.

A common component, containing the "K" adjustment as illustrated, is the base around which each Republic Null-Balance-Vector instrument is built. This common "heart" means that you have interchangeability of parts—even among instruments performing different functions. Conversion and replacement parts can be pooled instead of stocking spare instruments for each control function.

In addition, you benefit from simplified operating and maintenance training because the same principle of operation is used for all instruments. All Republic Vector instruments are compact, lightweight and rugged. The Republic Engineer in your area will be glad to work with you on any control or measurement application. Or, write for information to Republic Flow Meters Company, Subsidiary of Rockwell Manufacturing Company, 2240 Diversey Parkway, Chicago 47, Illinois. In Canada: Republic Flow Meters Canada Ltd., Toronto.

REPUBLIC INSTRUMENTS AND CONTROLS

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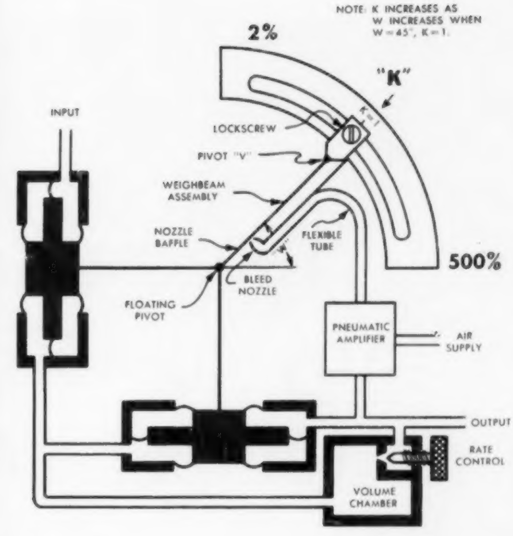
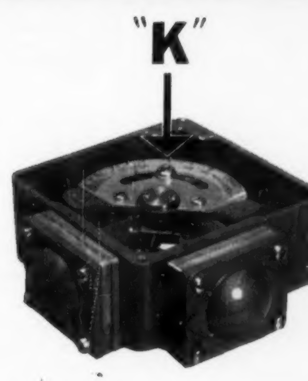
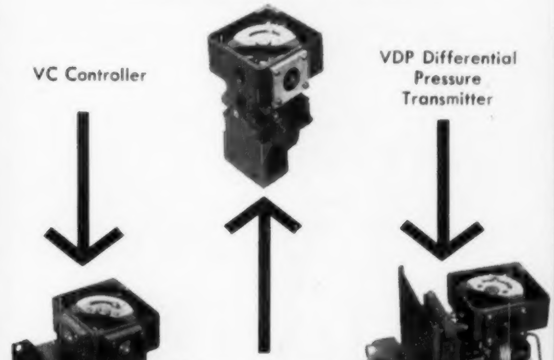
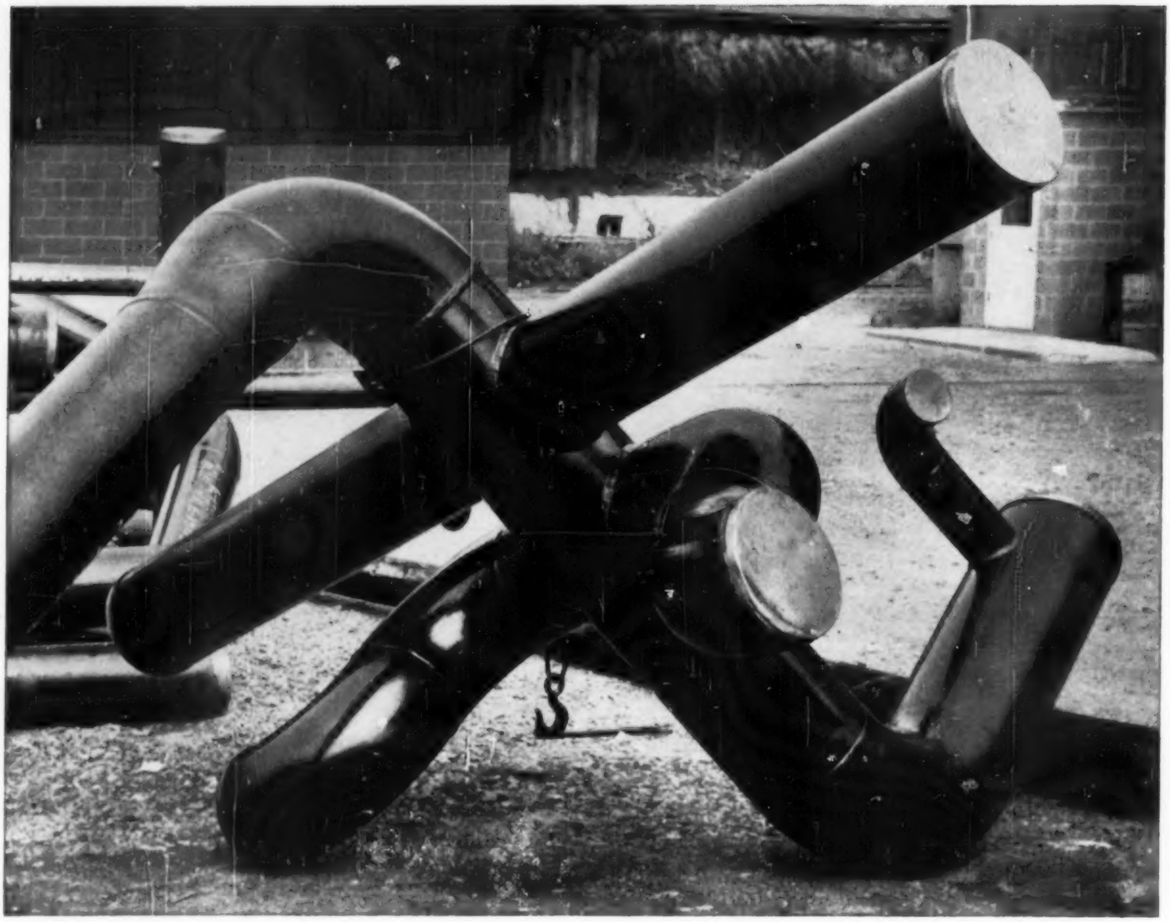



This is the common component around which each Republic Null-Balance-Vector pneumatic instrument is built. This component contains the "K" adjustment.

### TYPICAL REPUBLIC NULL-BALANCE-VECTOR INSTRUMENTS





Shop fabrication of complex shapes such as this 3/8" wall service water header simplifies field erection.

# Minimum field welding and fitting means faster installation and **lower overall cost of installed piping**

The service water header shown is part of an order which included all piping six inches and larger for a 325MW central station. The boiler will operate at 2000 psi and 1050-1000F. Main steam header was fabricated from 2¼ chrome, 1 moly pipe, 19" O.D. x 3½" wall.

Careful engineering, pre-planning and fabrication to high standards are the basic reasons why this job was erected with field welding and on-the-spot fitting held to a time-saving minimum. Contractor and owner both benefited from the ease with which the piping went into place—both saved money on the overall cost.

Complete facilities, engineering service, accurate fabrication that is quality control checked at every stage, plus erection service if desired . . . these are available to you on your next job. Take advantage of the *complete* piping services offered by Dravo. For full information, write DRAVO CORPORATION, PITTSBURGH 22, PENNSYLVANIA.

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# General and Classified Index

## COMBUSTION, Volume Thirty One, July 1959 Through June 1960

Page

### EDITORIALS



### AUTHORS

## CLASSIFIED

### Air Preheaters



### Boilers



### Coal and Ash Handling Systems



### Computers



### Condensers

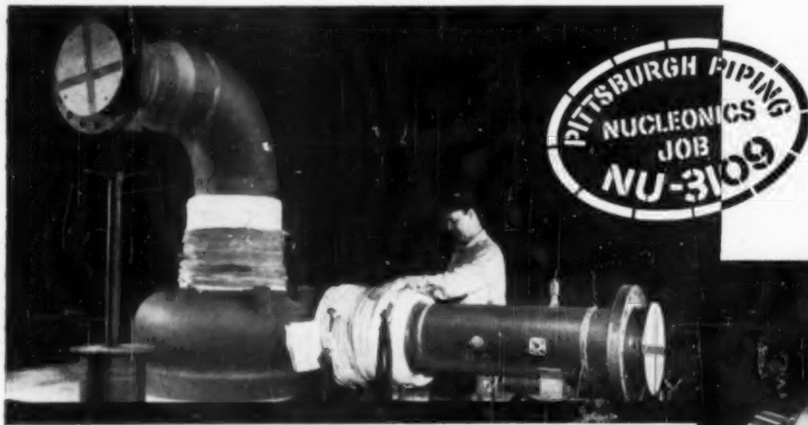


### Controls

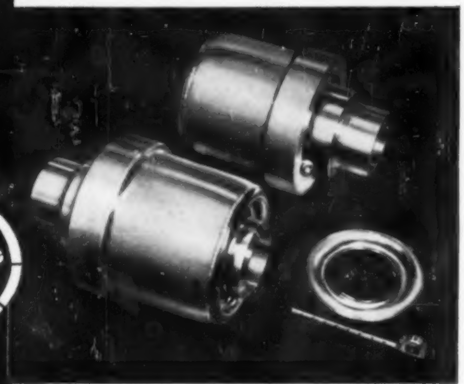
# Advertisers' Index



*(Continued on page 79)*

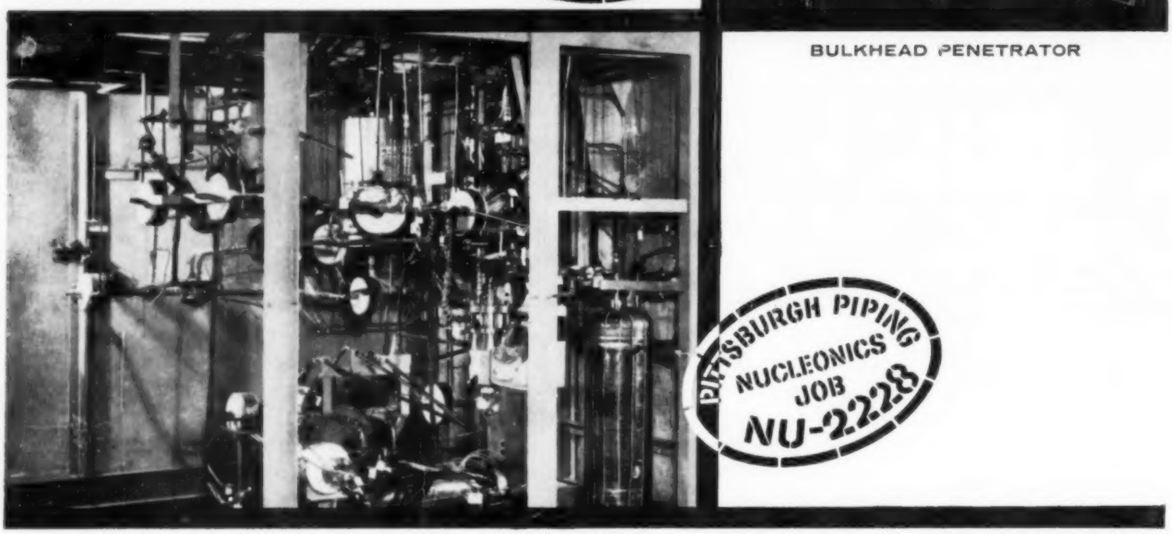



VOLUTE ASSEMBLY




BULKHEAD PENETRATOR

STAINLESS STEEL LOOP




# Fabricated by Pittsburgh Piping for Nucleonics

These jobs demonstrate the variety of nucleonic piping work done in our shops. The volute assembly, shown in intermediate stage of manufacture, is fabricated of 14" and 16" Schedule 160 piping materials. The bulkhead penetrator, one of a group, is notable for an unusual method of welding dissimilar metals. The complex test loop requires skillful welding of stainless steel piping materials, and precise assembly of hundreds of components. We are equipped and staffed to do similar jobs for you. Telephone, or write and our representative will call. Or, we will be glad to have you visit our plant.

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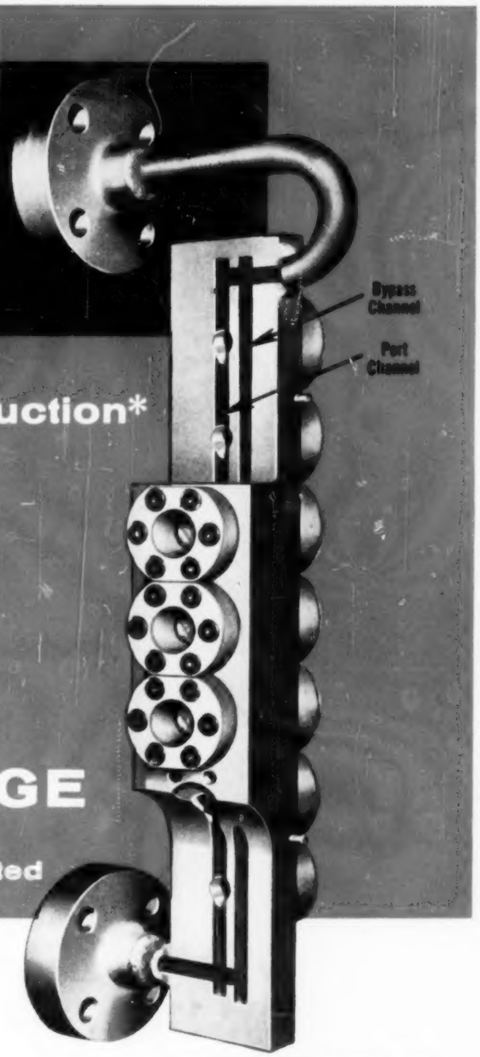


**New patented construction of the Diamond MP-3000 Multi-Port Bi-Color Gauge is illustrated by cut-away sections.**

This improved construction (exclusive with the Diamond MP-3000 Multi-Port Bi-Color Gauges) avoids possible confusion of colors in steam space of gauge and it improves visibility by providing sharper red and green colors.

Condensate, which might otherwise be a source of trouble, flows down the bypass channel instead of down the port channel. This permits the center plate to retain a relatively high temperature so that water in the ports is nearer to temperature of water in the boiler drum, improving indication accuracy. Absence of condensate in the ports assures a distinct Bi-Color image—signal red for steam—signal green for water.

Multi-Port construction greatly reduces gauge maintenance costs. Inherent thermal stability minimizes maintenance frequency and any port can be changed in a few minutes without removing the gauge from the boiler. Change of any port is expedited by new glass and gasket package unit. Construction is exceptionally sturdy—gauge body is machined from a solid block of high grade stainless steel. Get in touch with your local Diamond office, or write us at Lancaster, Ohio, P.O. Box 415, for complete information and bulletins.

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